THE PISTON MADE OF AIR

4th Edition

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The Piston Made of Air

Introduction

This book is about the Kadenacy Effect. This is the second book in my series on Maxwell's Demon. The first book, Maxwell's Demon Gets A Job, contains four chapters that I've written to fill in the gaps in the textbooks on compressed air, with a collection of supporting documentation. The Piston Made of Air contains sections documenting Kadenacy Engine theory, pulsejet engine theory, and pulsed-jet pump theory. Further volumes such as Resonance and Related Hardware, Vibration for Fluid Power Production, Energy Exchanges in Vibrating Fluids, Energy Exchanges in Vibrating Fluids, Constatinesco Treatise and Patents, Acoustic Power Patents of Albert G. Bodine, Jr. and Two-Stroke Exhaust Systems will document these same fluid wave phenomena that make it possible to design self-fueling air cars. There are many more-or-less well-known devices such as water ram pumps and cryptosteady pressure exchangers that use fluid waves to do the pumping work. (In engineering, a fluid is anything that flows, liquid or gas.)

In 1670, Christiaan Huyghens devised the first-ever engine, made up of a cannonball acting as a piston in a vertical cylinder. An explosion of gunpowder under the ball would raise it in the vertical cylinder, then it would fall back down the cylinder under its own weight. Huyghens' engine was able to pump water by the suddenness of the motion imparted to the cylinder's contents by the explosion. Behind the high pressure explosion pushing the ball up the cylinder, there was an implosion, a sudden and short-lived partial vacuum or rarefaction wave, in the cylinder. Because of the subatmospheric pressure, or depression, in the cylinder, the atmosphere added its own energy to this engine cycle to force the next cylinderful of water into the rarefied space under the ball. Then the ball fell back down the cylinder, pumping the water back out. Each explosion would "scavenge" the cylinder by clearing it of any water left over from the last cycle, by means of a high pressure blast or compression wave.

Huyghens' basic discovery--the depression left inside a closed cylinder after a sudden outward pulse--was the working principle that Thomas Savery used in taking the engine to the next stage in its development. The alternating wave in the fluid to be pumped comprised a fluid piston that did the pumping. Pulses of steam alternately scavenged the two chambers of Savery's engine, and each chamber would automatically refill with water because of the depression left behind each scavenging pulse. Savery's engine had no moving parts except valves. The mass exit of the chamber's contents left a depression that induced the next chamberful of water, and a pulse of steam pumped this water out. The pulsometer pump, which was manufactured from 1876 to at least 1938, used the same principle.

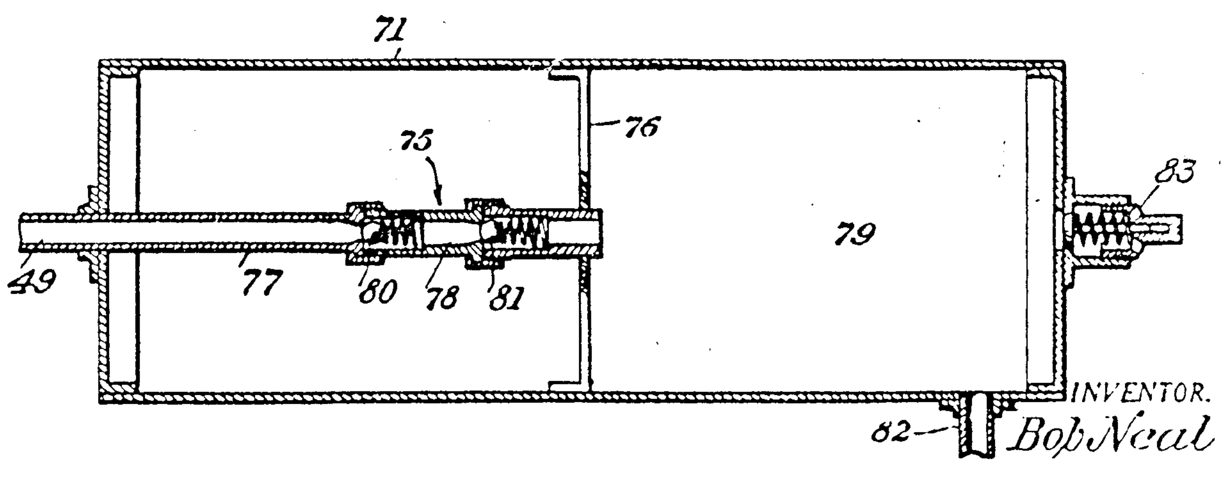
Newcomen's and Watt's cumbersome piston engines, laden wth expensive moving parts, took precedence in the developing engine industry over the simpler fluid piston principles that Huyghens and Savery pioneered. This was, in effect, an early version of engineered obsolescence.

Bob Neal's patented Compressor Unit is the best device I know of that can provide the essential hardware needed to run a self-fueling air car. Neal filed his patent in 1934; a trip to Washington with the working model secured him a patent in 1936; shortly thereafter he had to

abandon the project due to harassment by the Nazis. The Nazis perfected the pulsejet engine in 1943; the French were developing their own pulsejet, which had no valves, at the same time. This pistonless piston engine--a tube containing a self-sustaining fluid piston--proved to be the most powerful engine for its size that was ever built. And none of the pulsejet's inherent defects, such as high noise levels or wasted residual energy in the exhaust, apply to Neal's "equalizer" since the equalizer is inside a tank full of compressed air.

Michel Kadenacy filed the first of his twelve U.S. patents in 1934, eight months after Bob Neal filed his patent. Kadenacy filed his French patents on August 1, 1933, a few months after Hitler took power. Kadenacy's system of acoustically scavenging and charging an engine cylinder is still the theory behind two-stroke engine tuning. The Kadenacy Effect could be called the Huyghens Effect, but Huyghens already had a principle named after him. The Huyghens Principle describes the tendency of each wavelet in a wave field to propagate its own wave field. The work of Huyghens became the foundation for the work of James Clerk Maxwell, the founder of mathematical prediction in theoretical physics, who predicted the discovery of overunity dynamic pressure exchangers (Maxwell's Demon) in his textbook Theory Of Heat in 1870.

Arkansas shoemaker Bob Neal's compressor unit is Maxwell's Demon reduced to hardware, and it's also the logical idealization of Huyghens' first-engine-in-history. The Neal Equalizer will one day be considered a refinement in engineering thought that paved the way for the age of sustainable technology to take hold in the 21st century.



ABOVE: The Neal Equalizer
For more information see my book, *Maxwell's Demon Gets a Job*, or write Pneumatic Options, 1430 Willamette Street #561, Eugene, OR 97401, 541-683-4401.
Send $3.00 for my 40-page catalog of Pneumatic Options!

1. Kadenacy Effect

By 1934, many patents had been grantedfor air induction and scavenging by exhaust discharge pulse, according to P.W. Petter who began such experiments in 1922. A two-stroke engine he was working on unexpectedly started to cycle backwards, taking air in through the exhaust pipe and discharging it through the intake ports. "Remarkable practical results" came in about 1933 for Petter whose Superscavenge engines were among the first to commercially exploit the Kadenacy effect.

Michel Kadenacy's 12 U.S. Patents were granted between 1937 and 1939. "Kadenacy's basic contribution is his discovery that, immediately on sudden opening of exhaust ports during the expansion stroke, there is during the first few thousandths of a second thereafter, a rapid exhaust impulse that leaves behind a depression (subatmospheric pressure zone) in the cylinder. No scavenging pumps, inlet pipes or exhaust pipes are needed for such an engine to run and deliver power." (S.J. Davies, Engineering, 6-18-37)

The main use of the Kadenacy Effect is to replace other means of refilling the cylinder with fresh fuel/air mixture. The other methods are crankcase scavenging, and an independent blower. Scavenging refers to removal of exhaust gases from the cylinder. "The range taken, however, from 500 to 2000 rpm, shows a speed flexibility which is a remarkable achievement in an engine in which the only means of exhausting the burnt gases and of taking in a new charge are the simple ones described." (S. J. Davies, Engineering, 6-25-37).

"The extent of the vacuum created is largely dependent upon the force of the explosion(exhaust pulse)...because it leads to an increase in the volume of air admitted to the cylinder as the power demand upon the engine increases....Proper proportioning of the exhaust pipes is necessary in order to secure the best results." (P. W. Petter, The Engineer, 8-17-34)

Because of the intensity of the vacuum, on an experimental engine whose piston rod falls away freely after a power stroke, S.J. Davies reports, "The port was opened by the piston near the bottom of its stroke, the gases escaped, and it was observed that the piston returned up the cylinder and came to rest at a point about two-thirds of its upward travel." In another experiment, "56% of the original contents of the cylinder passed out through the exhaust port at a speed vastly higher than those accepted as possible in engine practice, and that, following this rapid exit, an equivalent depression of over 8 psi was left in the cylinder."

Davies' pioneering research confirmed Kadenacy's discovery that at a higher effective cylinder pressure, the scavenging and induction characteristics of a Kadenacy engine are better than at lower mean effective pressures. The more gases are in the cylinder, the easier it is to get them outand get new air in, using the Kadenacy Effect. Naturally this is the opposite of an engine not using the Kadenacy system of scavenging and induction. "...the volumetric efficiency is more than doubled from no load to full. This characteristic is brought about by the increase in the ballistic effects of the exhaust with increase of load." (S. J. Davies, Engineering, 6-18 & 25-89)

"If gases contained under pressure in a vessel are suddenly discharged by the rapid opening of a suitable orifice in the vessel, an explosion (exhaust pulse) of the gases from the vessel occurs, a

characteristic of which is that the pressure inside the vessel falls considerably below atmospheric and is followed by an implosion of the gases or air outside the vessel towards the interior, if the orifice remains open. Explosion and implosion, as far as their intensities are concerned, will be maxima when no pipe is fitted on the exhaust orifice....As the time interval between explosion and implosion is of the order of a few thousandths of a second, special arrangements have to be made to detect it." (James B. Henderson, Engineering, 9-22-39)

"Dr. Mucklow suggests that a depression cannot be created in the cylinder unless there is an exhaust pipe, because with no pipe the energy of the very short jet of gas at the port is insufficient to account for a big drop in pressure. This would be true for a large cylinder with a small orifice, but in the case I have quoted the port is so big that a comparatively large volume of the gas inside the cylinder would attain a high velocity, and besides this, the gas which has gone through the port will have sent off a positive pressure wave, leaving a negative wave at and around the port. Whatever explanation may be ultimately agreed on, there seems to be no doubt that engines built on the Kadenacy design do work remarkably well." (John C. Morrison, Engineering, September 29, 1939)

"...in some cases the assumptions underlying a theory are forgotten and the theory is applied to problems in which the assumptions do not hold. Such a case has arisen in the treatment of the exhaust of an internal combustion engine, as given in text-books on the subject. There is no branch of engineering science which lags farther behind practice than in problems involving the dynamics of fluid motion....for high velocities, such as arise in the motion of projectiles, the science is mostly empirical, the same applies to phenomena of the nature of explosions in the air. It is possible to calculate the maximum pressure due to say, the detonation of explosive against an armour plate, but the resulting motion produced in the air is beyond the powers of mathematical science....Textbook theory ignores what is happening outside the exhaust valve of an engine...this formula is based upon the assumption of a steady state in the vessel, so that the distribution of energy within the vessel may be ignored and the kinetic energy in the issuing jet equated to the work done in maintaining the constant pressure in the vessel...this formula gives results agreeing with experiment...when applied to conditions of a steady state....The exhaust of an engine is a transient phenomena; it is all over in a few thousandths of a second. Hence a great deal of justification is required in applying to it a formula refering it to a steady state...progressive wave phenomena are excluded. The only justification which can be given is that there is no other theoretical basis for a solution....We shall see later that the application of the above theory has only hampered progress....Kadenacy makes use, for the first time, in a two-stroke cycle engine, of the rarefaction in the cylinder following the explosive discharge of the exhaust to suck in the fresh charge, and succeeds in supercharging to such an extent that he can eliminate the charging compressor and obtain a larger horsepower than was obtained with it." (Professor Sir James B. Henderson, D.Sc., LL.D., 10-6-39, Engineering)

"...on the Kadenacy scavenging system, some experiments were devised to demonstrate...that a depression is created in a vessel after the sudden discharge of a gas under pressure through a large orifice, without the help of a discharge pipe. It would appear that the existence of such a depression follows from Newton's 3rd Law alone, without reference to any other theory or authority. The law states that the rate of change of momentum is equal to the force applied. During the discharge a mass of gas is moving out of the vessel representing a certain momentum in an outward direction. This momentum must be dissipated before equilibrium conditions are established, and that can only be done by an opposing force. If the orifice is relatively small, damping by friction provides this force, but if the orifice is large the gas can only be stopped and its momentum dissipated by an excess of pressure outside the vessel over the pressure inside it, representing the requisite force. Hence, a depression must be created in the container....The addition of an exit pipe on to the orifice adds some "inertance" to the system and its effect is comparable to that of an additional inductance in an electrical oscillating circuit. It alters the timing of the whole process of discharge, but not its nature...these simple experiments prove "conclusively" the much discussed depression after rapid discharge from a vessel, quite independent from the existence of a pipe, and that M. Kadenacy's statement to this effect, which originally started the controversy, was justified." (S.G. Bauer, Engineering, 2-2-40)

"Whether the vacuum is produced by the exhaust gases escaping en masse from the cylinder and leaving behind them a depression, or whether it is solely due to the negative reflection of a pressure pulse from the open end of a pipe, is of secondary importance... Whatever the effect of the exhaust pipe length on the magnitude of the depression, it seems certain that it can exert a powerful influence both on the duration of the "vacuum period" and on the interval between the point of opening of the exhaust valve and the instant when the pressure in the cylinder falls below atmospheric." (L. J. Kastner, Engineering, 2-23-40)

"Mr. Kastner asserts that the method of producing the depression in the cylinder...is of secondary importance. In an engine working on the Kadenacy system, this question is of primary importance, since, although pressure waves may be present in the pipe, the exhausting en masse dominates and determines the function of the exhaust pipe... The high momentum of these gases, both within the cylinder and outside, that is, in the exhaust organ or beyond it, should be considered." (S. J. Davies, Engineering, 2-23-40)

"...a further feature of the Kadenacy principle is the control of the motion of the exhaust gases after they have left the cylinder, which is accomplished by so arranging the exhaust system so that the resistance offered to the outgoing mass of burnt gases is reduced to a minimum, and the mass of gas is kept moving in an outward direction a sufficiently long time, in order to enable the cylinder to be recharged with incoming air...the successful operation of the engine with natural aspiration...does not preclude the addition of a blower, which may be utilised either for stabilisation or for supercharging the engine. In the case of the Kadenacy engine, however, the power required to drive the blower...is less than that for an engine that does not possess the natural aspiration or suction effect associated

with the Kadenacy process...the scavenge pump was rendered inoperative and the engine operated with natural aspiration...when converted to the Kadenacy system and run under conditions of maximum rating, the power range was increased...the power output was increased by 130%...the fuel consumption was reduced from .46 to .36 lb....when operating with natural aspiration, the engine draws more than one cylinder volume of fresh air into the cylinder. With the blower in action, about two cylinder volumes per cycle are passed, the pressure between the blower and the cylinder only amounting to about 2 psi...and these special tests were made with thirteen different lengths of tail pipe, ranging from nothing up to 150 feet. The results show that throughout these tests the performance of the engine was not materially affected...Another fact noted from the tests was the way in which the changing pressure of the engine fell off with an increase of the brake mean effective pressure. This reduced charging pressure, it is held, is explained by the fact that in the Kadenacy system the greater the energy in the exhaust, the greater will be the suction effect, tending to draw air into the cylinder, and thereby reducing correspondingly the resistance to the delivery of the air from the blower." (<u>The Engineer</u>, 3-8-40)

"...Whether the discharge process is visualised as distinct waves traveling through the discharge and being reflected at the end with a reversal of sign or as a surging movement of the whole mass of gas, the duration of the depression must always be related to the length of the system; that is to say, the combined length of the vessel and the discharge pipe, if any...the assumption, with which Dr. Davies started this controversy, that only the duration, but not the magnitude of the depression created by the sudden discharge from a vessel, is influenced by the existence and legth of a discharge pipe...the exhaust valve is opened suddenly, and the pressure or potential energy of the combustion products is transformed into kinetic energy...the exhaust gases, moving by now at a high velocity, draw air through the inlet into the cylinder, rather like a rapidly moving piston, and farther on into the exhaust pipe, if there is one... If this cycle is achieved no vacuum is actually created at all, and only if the inlet valve were not opened would the case be similar to that of the discharge vessel where the kinetic energy of the escaping gases is again transformed into potential energy by creating a depression. The magnitude of this depression is thus merely a measure of the energy which would have been available for scavenging purposes had an inlet valve been opened at the correct moment." (S. G. Bauer, <u>Engineering</u>, 4-26-40)

2. Pulsejet

"The general public impression has been that the pulsejets are purely an implement of war, but Ford engineers and others think differently. They are such an effective propellant--producing an estimated three horsepower for each pound of the more than 300 pound weight--that they have stimulated the imagination of transportation engineers everywhere. They are being thought of as a propellant for aircraft, marine vessels and automobiles. The principle of the impulse duct engine borne as a desparate war measure in the minds of the German engineers may find its fullest destiny as an instrument of peace." (S. H. Brams, <u>The Iron Age</u>, 2-1-45)

"Watching the testing of one of the (pulsejet) engines on the test stand at Rouge is a soul-shaking experience. The observer feels that he is in the presence of power greater than anything he has ever experienced. In order to start the engine, a blast of (compressed) air is directed at the grill with sufficient velocity to duplicate the ram pressure attained in flight. As the engine roars into high speed, the shock waves of the successive explosions hit the observer with a physical force that can hardly be described. The vibrations are most disturbing physically and psychologically. Standing toward the rear opposite the flaming column of exhaust is as much as your life is worth. The writer stood it for several seconds and then walked away while he was still in one piece." (Aero Digest, 5-1-45)

"In the pulsejet engine each pound of gas uses probably a little more than thirteen pounds of air. Accordingly, less than one fourteenth of the weight of the jet comes out of the (gasoline) storage tanks of the vehicle...In the reaction motor there is no machine compression. Still, the air is compressed to some degree. The manner in which that is accomplished without compressor, or cylinder and piston assembly, is the novel feature. The air is made to compress itself for each new explosion as an aftermath of the previous explosion. The exhaust stack is dimensioned, and the system tuned, for the appearance of waves of high pressure. There is resonance between the succeeding explosions and the natural period of such pressure waves. Thus, essentially moving parts are done away with, and still compression is provided, apparently from nowhere. It appears that the exhaust stacks of the pulsejet are of considerable length, about eleven feet. A sound wave would require one fiftieth of a second to travel such a tube both ways. It would oscillate about 3000 times per minute, about as frequently as the pistons in modern gasoline engines. The stacks have one-way valve entrances in front... The vacuum itself...sucks air into the rear opening of the stack, the exit serving momentarily as an inlet. This backflow strikes the front of the stack, and a high impact pressure is built up by way of stopping the forward surge of the air. This is the compression effect relied on. The pressure lasts only a small fraction of the total period, which is itself not much longer than one fiftieth of a second. At that point the next explosion is timed to take place. A new cycle begins, the stack acting like an organ pipe, and the explosions being in resonance with the natural frequency of the explosion wave, and of the forward surge wave. It must have been difficult and time-absorbing to make such a scheme work. Even now it does not work well and is extremely inefficient. The wonder is that it works at all." (Dr. Max M. Munk, Aero Digest, 9-1-44)

"The cycle is not controlled by the valves but is the result of resonance in the tube, and of the speed of burning of the mixture. The valves are actuated automatically by pressure reversals in the tube." (Blaine Stubblefield, Aviation News, 2-12-45)

"An American-designed impulse-jet engine, stated to surpass that used on the Nazi V-1, has been developed by the G. M. Giannini Co., of Pasadena, Calif....Efficiency of the diminutive power plant is said to be so great that if a fuel tank were directly attached and the engine started off on an inclined plane it would continue to fly through the air, even without wings, until the fuel was exhausted. Specifically, company engineers claim greater efficiency for the engine over the German type because of more thrust, less weight, and smaller size. More power is developed because the American engine fires more

than five times faster than the German unit. The wind tunnel model, for instance, fires approximately 250 explosions per second, when operating at full throttle. (Develops 2 lb-thrust; 2 ft long; 2 inch inside diameter.) (Aviation, 10-45)

"...inventors William L. Tenney and Charles Marks of Minneapolis, have shrunk the (pulsejet) engine to toy size. For its measurements, it probably is the most powerful package of energy ever produced. It will drive a miniature automobile, for example, 150 mph...the 21 inch baby jet supplies a thrust of three pounds when not moving, more than that when movement rams air through the spring shutters in its nose...In larger sizes, the inventor believes, this "Dyna-jet" could be used to drive full-scale airplanes, racing cars and boats... The entire assembly weighs only a pound... The baby jet pulses up to 250 times a second." (Popular Science, 5-46)

"However, its efficiency is low, and at present its operation is limited to subsonic speeds because of shock wave interference at the inlet valves for supersonic-flow conditions. The device is not capable of sudden changes in power output or altitude and thus the maneuverability of the airframe it is installed in is limited." (William P. Munger, Aero Digest, 3-48)

Energy Content of High-Pressure Gases.

By E. W. GEYER, B.Sc., Ph.D.*

THE problem of determining the available energy from gases contained in closed vessels is of importance in the constant volume type of gas turbine. It also arises, however, in other cases, such as in the storage of high-pressure air used for starting oil engines and in the emptying of submarine water ballast tanks. Two cases are considered in this article, one without and the other with heat flow to the containing vessel.

AVAILABLE ENERGY WITH NO HEAT FLOW.

(a) *Complete Emptying.*—Let a vessel A, Fig. 1, of capacity V cubic feet contain gas at the absolute pressure p_1 and absolute temperature T_1, and let the external absolute pressure be p_b. In order to simplify the

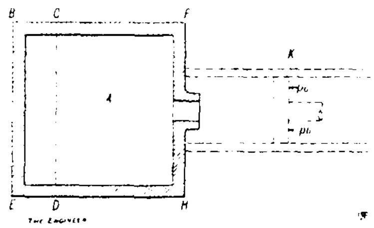


FIG. 1

theoretical consideration of this problem, it is convenient to imagine that the portion of the high-pressure gas, contained in the space B C D E expands down to the outer pressure p_b within the vessel itself, while the portion of the gas contained in the space C F H D is discharged through the nozzle. It can be assumed that this latter portion, on emerging from the nozzle, pushes an imaginary piston on the outer surface of which the constant external pressure p_b is exerted. Let the weights of the gas in the portions B C D E and C F H D be W_1 and W_2 lb. respectively, and let the total weight be W lb., *i.e.*, $W_1+W_2=W$. The internal energy of the gas in the vessel before discharge is $W I_1$, where I_1 is the internal energy per pound of gas at the temperature T_1. Since it is assumed that there are no heat nor friction losses, the expansion of the gas, both in the vessel and through the nozzle, is adiabatic, so that the final temperature T_2 in the vessel is the same as that in the imaginary cylinder K. If the internal energy per pound of gas at T_2 is denoted by I_2, the final store of internal energy in the vessel is $W_1 I_2$ and in the imaginary cylinder $W_2 I_2$. The total final store of internal energy is thus $(W_1+W_2)I_2 = W I_2$. Before the energy law can be expressed, it is necessary to determine the external work performed on the imaginary piston. Let v_2 be the specific

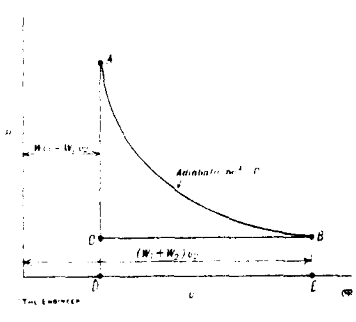


FIG. 2

volume of the gas after expansion. Then, since the external pressure exerted on the piston is constant and equal to p_b, the work done by the gases on this piston is $144\ p_b v_2 W_2$ ft. lb., or in heat units $144\ p_b v_2 W_2/J$.

The energy law now gives the available work as

$$\frac{E}{J}=W I_1-W I_2-\frac{144\ p_b v_2 W_2}{J}$$

$$=W(I_1-I_2)-\frac{144\ p_b v_2 W_2}{J} \qquad \ldots \quad (1)$$

This work can be represented on the pressure volume field as follows :—Let A, Fig. 2, represent the state of the gas in the vessel at the beginning of discharge, and let B represent the state of the gas, both in the vessel and in the imaginary cylinder at the end of discharge. The curve A B represents the adiabatic expansion of the gas, both in the vessel and through the nozzle. The area A B E D below the curve A B represents the difference of internal energy $W(I_1-I_2)$ before and after the adiabatic expansion, and this expression appears as the first term on the right of equation (1). The distance $C B = W_1 v_2 + W_2 v_2 - W_1 v_2 = W_2 v_2$, represents the volume swept out by the imaginary piston. The area C B E D is thus equal to $144\ p_b v_2 W_2/J$, and this appears as the second term on the right of equation (1). Hence, equation (1) is represented by the difference of the areas A B E D and C B E D in Fig. 2, *i.e.*, the available work of expansion with no friction and no heat flow is represented by the area A B C.

(b) *Partial Emptying.*—If the discharge of gas from the vessel ceases by closing the valve when the internal pressure is above that of the external pressure, the available energy and its representation on the P V field are obtained as follows :—

As before, let W_1 be the weight of gas which expands and remains in the vessel itself, and let W_2 be the weight which is discharged through the nozzle. Also let the pressure in the vessel, at the instant the discharge through the nozzle ceases, be p_2. The total internal energy in the vessel initially is $(W_1+W_2) I_1$. Let T_2 and T_3 be the temperatures of the gas after expansion in the vessel, and in the imaginary cylinder respectively, and let I_2 and I_3 be the corresponding internal energies per pound. The final store of internal energy is thus $(W_1 I_2+W_2 I_3)$. If the final specific volume in the imaginary cylinder is denoted by v_3 the work in heat units performed by the W_2 lb. of gas against the external pressure p_b is $\frac{144\ W_2 v_3 p_b}{J}$. The available energy is now given by the decrease in internal energy less the external work performed, *i.e.*,

$$\frac{E}{J}=(W_1+W_2)I_1-W_1 I_2-W_2 I_3-\frac{144\ W_2 v_3 p_b}{J}$$

$$=(W_1+W_2)(I_1-I_2)+W_2(I_2-I_3)-\frac{144\ W_2 v_3 p_b}{J}$$

But $W_1+W_2=W$, and $W_2=W-W_1$, so that

$$\frac{E}{J}=W(I_1-I_3)-W_1(I_2-I_3)$$

$$-\frac{144\ W_2 v_3 p_b}{J} \qquad \ldots \ldots \quad (2)$$

This is represented on the p V field as follows :—Let A, Fig. 3, represent the state point of the gas in the vessel before discharge and let the gas pressure drop

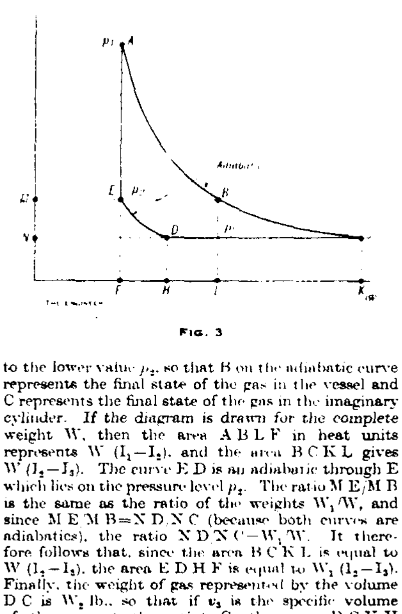


FIG. 3

to the lower value p_2, so that B on the adiabatic curve represents the final state of the gas in the vessel and C represents the final state of the gas in the imaginary cylinder. If the diagram is drawn for the complete weight W, then the area A B L F in heat units represents $W(I_1-I_2)$, and the area B C K L gives $W(I_2-I_3)$. The curve E D is an adiabatic through E which lies on the pressure level p_2. The ratio M E/M B is the same as the ratio of the weights W_1/W, and since M E/M B = N D/N C (because both curves are adiabatics), the ratio N D/N C = W_1/W. It therefore follows that, since the area B C K L is equal to $W(I_2-I_3)$, the area E D H F is equal to $W_1(I_2-I_3)$. Finally, the weight of gas represented by the volume D C is W_2 lb., so that if v_3 is the specific volume of the gas at the point C, the area D C K H $=144\ p_b W_2 v_3/J$ heat units.

Hence $\frac{E}{J}$, which is given above in equation (2), is represented on the p V field by the following areas :—

$$E/J = ABLF + LBCK - EDHF - DCKH$$
$$= ACDE.$$

The following example serves to show the application of this treatment :—Air at a pressure of 114·8 lb. per square inch abs. and a temperature of 297 deg. Cent. abs. is discharged from a vessel having a capacity of 2·95 cubic feet through a nozzle into the atmosphere. Find the ideal available jet energy if the pressure in the vessel drops to the external pressure of 14·8 lb. per square inch abs. Also show the available jet energies if the discharge ceases when the internal pressure drops to the following values :— $p_2=100, 90, 80, 70, 60, 50, 40, 30$, and 20 lb. per square inch abs. Take $\gamma=1\cdot4$ and $C_v=0\cdot169$.

With complete emptying the pressure in the vessel drops to $p_b=14\cdot8$ lb. per square inch abs., so that the final temperature both before and after the nozzle is

$$T_2=\frac{T_1}{\left(\frac{p_1}{p_b}\right)^{\frac{\gamma-1}{\gamma}}}=\frac{297}{\left(\frac{114\cdot8}{14\cdot8}\right)^{\frac{0\cdot4}{1\cdot4}}}=165\cdot6 \text{ deg. Cent. abs.}$$

The initial weight of air in the vessel is

$$W=144\ p_1 V/R\ T_1=\frac{144\times114\cdot8\times2\cdot95}{96\cdot3\times297}=1\cdot71 \text{ lb.}$$

The final weight of air in the vessel is

$$W_1=\frac{144\ p_b V}{R\ T_2}=\frac{144\times14\cdot8\times2\cdot95}{96\cdot3\times165\cdot6}=0\cdot394 \text{ lb.}$$

Hence the weight of air discharged is

$$W_2=W-W_1=1\cdot71-0\cdot394=1\cdot316 \text{ lb.}$$

The final specific volume is given by

$$v_2=\frac{R\ T_2}{144\ p_b}=\frac{96\cdot3\times165\cdot6}{144\times14\cdot8}=7\cdot49 \text{ ft.}^3/\text{lb.}$$

The available jet energy is now found from equation (1), *i.e.*,

$$\frac{E}{J}=W(I_1-I_2)-\frac{144\ p_b v_2 W_2}{J}$$

or, since the expansion occurs at low temperatures, for which the specific heat C_v of the gas at constant volume may be taken as constant,

$$\frac{E}{J}=W C_v(T_1-T_2)-\frac{144\ p_b v_2 W_2}{J}$$

$$=1\cdot71\times0\cdot169(297-165\cdot6)-\frac{144\times14\cdot8\times7\cdot49\times1\cdot316}{1400}$$

$$=38\cdot0-15\cdot0=23\cdot0 \text{ C.H.U., or } 32{,}200 \text{ ft.-lb.}$$

In the case of partial emptying the available energy is given by equation (2), *i.e.*,

$$\frac{E}{J}=W(I_1-I_3)-W_1(I_2-I_3)-144\ p_b W_2 v_3/J$$

$$=W C_v(T_1-T_3)-W_1 C_v(T_2-T_3)-144\ p_b W_2 v_3/J.$$

Choosing a pressure drop in the vessel from $p_1=114\cdot8$ lb. per square inch abs. to $p_2=90$ lb. per square inch abs., the following figures are obtained.

The temperature at the end of expansion in the vessel is given by

$$T_2=\frac{T_1}{\left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}}=\frac{297}{\left(\frac{114\cdot8}{90}\right)^{\frac{0\cdot4}{1\cdot4}}}=277 \text{ deg. Cent. abs.}$$

The temperature T_3 at discharge from the nozzle is, as in the case of complete discharge, equal to 165·6 deg. Cent. abs.

The initial weight W in the vessel is the same as before, namely, 1·710 lb. The final weight W_1 in the vessel is given by

$$W_1=\frac{144\ p_2 V}{R\ T_2}=\frac{144\times90\times2\cdot95}{96\cdot3\times277}=1\cdot432 \text{ lb.}$$

The weight discharged is thus

$$W_2=W-W_1=1\cdot710-1\cdot432=0\cdot278 \text{ lb.}$$

The specific volume of the discharged gas is, as before, $v_3=7\cdot49$ ft.^{3}/lb. Hence

$$\frac{E}{J}=1\cdot710\times0\cdot169(297-165\cdot6)-1\cdot432$$

$$\times0\cdot169(277-165\cdot6)-\frac{144\times0\cdot278\times7\cdot49\times14\cdot8}{1400}$$

$$=7\cdot8 \text{ C.H.U., or } 10{,}920 \text{ ft.-lb.}$$

The available energies for the other final pressures are obtained in the same way and are shown in the table below :—

Final pressure in vessel p_2.	Available energy E/J C.H.U.	Available energy E, ft.-lb.	Percentage available energy.
100	4·9	6,860	21
90	7·8	10,920	34
80	10·9	15,260	47
70	13·6	19,040	59
60	16·0	22,400	70
50	18·4	25,760	80
40	20·4	28,560	89
30	22·0	30,800	96
20	22·9	32,060	99·6
14·8	23·0	32,200	100

Curves showing the above relationships graphically have been plotted in Fig. 4.

AVAILABLE ENERGY WITH HEAT FLOW.

It has been shown above how the available jet energy of a compressed gas in a vessel of constant volume can be found from the energy law when no heat flow occurs. The problem of determining the available energy when heat flows into the vessel during the discharge of the gas will now be discussed. The law of expansion of the gases in the vessel may be expressed by the simple relationship $pv^n=c$, where n is the mean index of expansion, while the law of expansion through a smooth well-rounded nozzle may be taken as following the law $pv^\gamma=c$, where γ is the adiabatic index of expansion. The work obtained by the assumption of an adiabatic expansion through the nozzle can be multiplied by a nozzle efficiency factor to give the actual available jet energy.

In Fig. 5 let A represent the state point of the gas in the vessel before discharge occurs. The curve A E is an adiabatic between the inlet pressure p_1 and the external pressure p_b. Let A D be the polytropic representing the expansion of the residual gases in the vessel. The kinetic energy of the first element of gas discharged through the nozzle is represented by the area E_1 = F A E K. At some later instant, when the pressure has dropped to p_2, the state of the gas in the vessel is represented by the point B, so that the energy of an element passing through the nozzle is given by the area H B C K, where B C is an adiabatic curve passing through B.

The weight of gas discharged between A and B is

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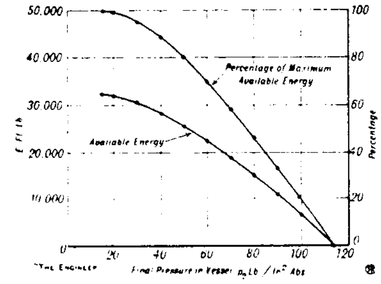

FIG. 4

$$W = \frac{V}{v_1} - \frac{V}{v_2} \quad . \quad . \quad . \quad (3)$$

where V is the volume of the vessel and v_1 and v_2 are the specific volumes of the gas in the vessel at A and B respectively.

Differentiating equation (3) with respect to v_2 gives

$$dW = -V\,d\left(\frac{1}{v_2}\right), \text{ or, since } \frac{1}{v_2} = \rho_2 = \text{density of the gas}$$

at the point B.

$$dW = -V\,d(\rho_2) \quad . \quad . \quad . \quad (4)$$

When the pressure in the vessel drops to p_2, i.e., to the point B, the jet energy per pound of an element of weight dW discharged through the nozzle is represented by the area H B C K, which, in heat units, is the same as the difference in total heats between the points B and C. The area H B C K is thus equal to $J C_p (T_2 - T_3)$ ft.-lb., and the jet energy of the element is

$$dE = J C_p (T_2 - T_3)\,dW.$$

or, from equation (4)

$$dE = -J C_p V (T_2 - T_3)\,d(\rho_2).$$

and the total available kinetic energy as the pressure in the vessel drops from p_1 to p_2 is

$$E = -\int_{\rho_1}^{\rho_2} J C_p V (T_2 - T_3)\,d(\rho_2)$$

If the expansion occurs at moderate temperatures C_p may be assumed constant, so that

$$E = -J C_p V \int_{\rho_1}^{\rho_2} (T_2 - T_3)\,d(\rho_2)$$

From the polytropic law for the curve A B we have

$$\frac{p_1}{\rho_1^n} = \frac{p_2}{\rho_2^n}$$

so that

$$\rho_2 = \rho_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

and

$$d(\rho_2) = \frac{\rho_1}{p_1^{1/n}} \times \frac{1}{n}\, p_2^{\frac{1}{n}-1}\, d(p_2).$$

hence

$$E = -J C_p V \int_{p_1}^{p_2} (T_2 - T_3) \frac{\rho_1}{p_1^{1/n}} \times \frac{1}{n}\, p_2^{(1/n - 1)}\, d(p_2)$$

or since $\rho_1 = \frac{1}{v_1}$

$$E = -\frac{J C_p V}{p_1^{1/n} v_1 n} \int_{p_1}^{p_2} (T_2 - T_3)\, p_2^{(1/n-1)}\, d(p_2) \quad (5)$$

Fig. 5 shows that T_2 is the temperature at B and T_3 the temperature at C, so that

$$T_2 = \frac{T_1}{\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}} \text{ and } T_3 = \frac{T_2}{\left(\frac{p_2}{p_b}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{T_1}{\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}} \left(\frac{p_2}{p_b}\right)^{\frac{\gamma-1}{\gamma}}}$$

so that

$$T_2 - T_3 = \frac{T_1}{p_1^{(1-1/n)}}\left[p_2^{(1-1/n)} - p_b^{(1-1/\gamma)}\, p_2^{(1/\gamma - 1/n)}\right]$$

Equation (5) thus becomes

$$E = -\frac{J C_p V T_1}{p_1^{1/n} v_1 n\, p_1^{(1-1/n)}} \times \int_{p_1}^{p_2} \left[p_2^{(1-1/n)} - p_b^{(1-1/\gamma)}\, p_2^{(1/\gamma-1/n)}\right] p_2^{(1/n-1)}\, d(p_2)$$

$$= -\frac{J C_p V T_1}{p_1 v_1 n}\int_{p_1}^{p_2}\left[1 - p_b^{(1-1/\gamma)}\, p_2^{(1/\gamma - 1)}\right] d(p_2)$$

which, on integrating, gives

$$E = \frac{J C_p V T_1}{p_1 v_1 n}\left[(p_1 - p_2) - \gamma\, p_b^{(1-1/\gamma)}\left(p_1^{1/\gamma} - p_2^{1/\gamma}\right)\right]$$

or, since $\frac{V}{v_1} = W$ = weight of the contents of the vessel before discharge,

$$E = \frac{J C_p W T_1}{p_1 n}\left[(p_1-p_2) - \gamma\, p_b^{(1-1/\gamma)}\left(p_1^{1/\gamma} - p_2^{1/\gamma}\right)\right] \quad (6)$$

It should be noted that in this expression, the pressures may either be in pounds per square inch

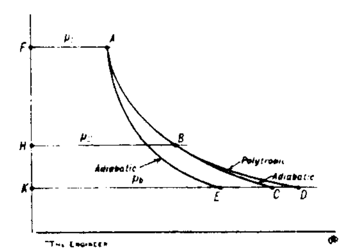

FIG. 5

abs. or in pounds per square foot abs., since the pressure dimensions cancel.

The following example shows the application of equation (6):—

Air at a pressure of 114·8 lb. per square inch abs. and temperature of 297 deg. Cent. abs. is discharged through a convergent nozzle into the atmosphere from a vessel having a constant capacity of 2·95 cubic feet. Find the total available jet energy if the pressure in the vessel is reduced during discharge to the external pressure of 14·8 lb. per square inch abs. Take the mean index of expansion of the air in the vessel as $n = 1\cdot05$.

Also show the available jet energies if the discharge ceases when the internal pressure p_2 drops to the following values:—p_2 = 100, 90, 80, 70, 60, 50, 40, 30, and 20 lb. per square inch abs., the corresponding

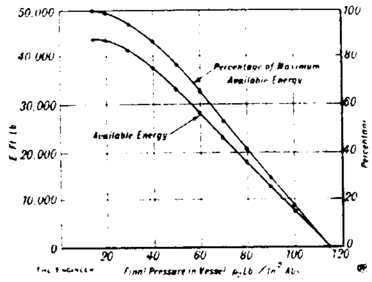

FIG. 6

n values being $n = 1\cdot26$, 1·21, 1·18, 1·15, 1·12, 1·09, 1·07, 1·06, and 1·05.

Assume adiabatic flow through the nozzle and take $C_p = 0\cdot238$.

The initial weight of air in the vessel is

$$W = \frac{144\, p_1 V}{R T_1} = \frac{144 \times 114\cdot8 \times 2\cdot95}{96\cdot3 \times 297} = 1\cdot71 \text{ lb.}$$

The available energy with complete discharge, i.e., with $p_2 = p_b$, is now given by equation (6), i.e.,

$$E = \frac{J C_p W T_1}{p_1 n}\left[(p_1 - p_2) - \gamma\, p_b^{(1-1/\gamma)}\left(p_1^{1/\gamma} - p_2^{1/\gamma}\right)\right]$$

$$= \frac{1400 \times 0\cdot238 \times 1\cdot71 \times 297}{114\cdot8 \times 1\cdot05}\left[(114\cdot8 - 14\cdot8) - 1\cdot4 \times 14\cdot8^{0\cdot4/1\cdot4}\left(114\cdot8^{1/1\cdot4} - 14\cdot8^{1/1\cdot4}\right)\right]$$

= 43,800 ft.-lb., or 31·1 C.H.U.

Dealing now with the case of partial emptying of the vessel and choosing a pressure drop to $p_2 = 90$ lb. per square inch abs., equation (6) gives

$$E = \frac{1400 \times 0\cdot238 \times 1\cdot71 \times 297}{114\cdot8 \times 1\cdot275}\left[(114\cdot8 - 90) - 1\cdot4 \times 14\cdot8^{1/3\cdot5}\left(114\cdot8^{1/1\cdot4} - 90^{1/1\cdot4}\right)\right]$$

= 12,300 ft.-lb., or 8·79 C.H.U.

The available jet energies for the other final pressures were calculated in the same way, and the values are tabulated below. Curves showing the relationship between p_2 and the percentage ratio of the available energy to the energy obtained with complete discharge, are shown in Fig. 6.

Final pressure in vessel = p_2.	Available energy E/J C.H.U.	Available energy E. ft.-lb.	Percentage available energy.
100	5·56	7,790	18
90	8·51	12,300	28
80	13·04	18,260	42
70	16·71	23,400	53
60	20·43	28,600	65
50	24·29	34,000	78
40	26·86	37,600	86
30	29·43	41,200	94
20	31·14	43,600	99·5
14·8	31·29	43,800	100

Time of Discharge of High-Pressure Gases

By E. W. GEYER, B.Sc., Ph.D.*

IN a previous article† the author discussed the problem of determining the available energy of high-pressure gases discharged through nozzles from a vessel into which no flow occurs. It is now proposed to deal with the question of the time taken for the gases to drop from an initial pressure p_1 to any lower pressure p_2.

Consider a vessel of volume V ft.³ containing gas at the absolute pressure p_1 and absolute temperature T_1 and let the corresponding specific volume be v_1. Due to the escape of gas through a nozzle, say, the pressure and generally the temperature drop to the lower values p_2 and T_2 for which the corresponding specific volume is v_2. At any intermediate instant let the pressure, temperature, and specific volume be p, T, and v.

The weight discharged in time dt is

$$dW = -V\,d\left(\frac{1}{v}\right) = -V\,d\rho \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where ρ is the density of the gas.

The instantaneous rate of mass discharge is given by

$$W = C_d A \sqrt{2g\,\frac{\gamma}{\gamma-1}\,\frac{p}{v}\left[\left(\frac{p_e}{p}\right)^{\frac{2}{\gamma}} - \left(\frac{p_e}{p}\right)^{\frac{\gamma+1}{\gamma}}\right]} \quad (2)$$

where C_d is the coefficient of discharge of the nozzle,
A is the area of the nozzle,
γ is the adiabatic expansion index,
p_e is the pressure at the immediate nozzle exit.

It should be noted that p_e will be the same as the pressure p_b beyond the nozzle exit if the ratio p_b/p is above the critical pressure ratio, but p_e will be greater than p_b if the ratio p_b/p is equal to the critical pressure ratio. It is thus seen that p_e/p has an important influence on the mass rate of discharge. Its critical value is given by $(p_e/p)_{cr} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$, the value of which is 0·528 for diatomic gases with $\gamma = 1\cdot4$. The instantaneous rate of mass discharge, when p_b/p is equal to, or less than, 0·528, is thus obtained by substituting the value 0·528 for p_e/p in equation (2), giving

$$W = 3\cdot89\, C_d A \sqrt{\frac{p}{v}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

By writing

$$\beta = \sqrt{2g\,\frac{\gamma}{\gamma-1}\left[\left(\frac{p_e}{p}\right)^{\frac{2}{\gamma}} - \left(\frac{p_e}{p}\right)^{\frac{\gamma+1}{\gamma}}\right]} \quad . \quad . \quad (4)$$

then, for all cases, i.e., whether or not p_e/p is constant, equation (2) gives

$$W = C_d A \beta \sqrt{\frac{p}{v}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

When p_e/p is equal to the critical value, β is constant, otherwise β is a function of p_e/p.

The weight of gas discharged in an element of time dt is thus

$$dW = C_d A \beta \sqrt{\frac{p}{v}}\,dt \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

so that with equation (1)

$$C_d A \beta \sqrt{p\rho}\,dt = -V\,d\rho.$$

If the law of expansion of the gases within the vessel be denoted by $p/\rho^n = p_1/\rho_1^n$,

then $p = p_1\left(\frac{\rho}{\rho_1}\right)^n$

so that

$$C_d A \beta \sqrt{\frac{p_1}{\rho_1^n} \times \rho^{n+1}}\,dt = -V\,d\rho,$$

giving

$$C_d A \beta\, p_1^{\frac{1}{2}}\, v_1^{\frac{n}{2}}\,dt = -V\rho^{-\left(\frac{n+1}{2}\right)}\,d\rho \quad . \quad (7)$$

This is the fundamental equation which enables the time of discharge to be determined. If this time is long compared with the times of opening and closing of the nozzle valve the area A may be assumed constant. If, however, this is not the case the variation of A with time has to be taken into account in integrating equation (7). For a nozzle which is well rounded at inlet and parallel beyond the throat, C_d may also be assumed constant,

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† "Energy Content of High-pressure Gases," THE ENGINEER, September 2nd, 1938.

otherwise its variation with the varying conditions of discharge must be taken into account. In the examples quoted below, which are taken from actual experiments, the time of opening and closing of a relatively large mushroom valve were very short compared with the total period of discharge, so that a constant area could safely be assumed and since the nozzles were short and of convergent-parallel design, C_d could be taken as constant and equal to 0·975.

It is convenient to refer to the times during which p_b/p remains constant at the critical value as the subcritical region and to the times during which p_b/p is above the critical value as the supercritical region.

Subcritical Region.—For the subcritical region β (equation (4)) is constant, so that, for the cases where A and C_d are also constant, equation (7) can be integrated directly. This gives

$$C_d A p_1^{\frac{1}{2}} v_1^{\frac{n}{2}} t = \left(\frac{2 V \rho^{\frac{n-1}{2}}}{n-1}\right)_{\rho_1}^{\rho_2} = \frac{2 V}{n-1}\left(v_2^{\frac{n-1}{2}} - v_1^{\frac{n-1}{2}}\right)$$

or

$$t = \frac{2 V v_1^{\frac{n-1}{2}}\left\{\left(\frac{v_2}{v_1}\right)^{\frac{n-1}{2}} - 1\right\}}{(n-1) C_d \beta A p_1^{\frac{1}{2}} v_1^{\frac{n}{2}}}$$

$$= \frac{2 V}{(n-1) C_d \beta A \sqrt{R T_1}}\left\{\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{2n}} - 1\right\} \quad (8)$$

The following example serves to show the application of equation (8).

Air at a pressure of 114·8 lb. per square inch absolute, and temperature of 297 deg. Cent. absolute is discharged from a vessel having a capacity of 2·95 cubic feet through a convergent parallel nozzle having a cross-sectional area of $2\cdot86 \times 10^{-4}$ square feet, and a coefficient of discharge of 0·98. Find the time required to drop the pressure in the vessel from the initial value to 28 lb. per square inch absolute. The external pressure is 14·8 lb. per square inch absolute, and the mean index of expansion, as determined experimentally, is $n = 1\cdot10$.

This expansion occurs entirely in the subcritical region, since the final pressure in the vessel is 28 lb. per square inch absolute, and the outer pressure is $p_b = 14\cdot8$ lb. per square inch absolute, giving $p_b/p = 14\cdot8/28 = 0\cdot528$. For all pressures above 28 lb. per square inch absolute, the ratio p_b/p is less than 0·528.

Equation (8) thus gives

$$t = \frac{2 \times 2\cdot95\left[\left(\frac{114\cdot8}{28\cdot0}\right)^{\frac{0\cdot10}{2\cdot20}} - 1\right]}{2\cdot86 \times 10^{-4} \times 0\cdot10 \times 0\cdot98 \times 3\cdot89\sqrt{96\cdot3 \times 297}}$$

$= 21\cdot2$ secs.

Supercritical Region.—In this region β is a function of p_b/p—see equation (4). The integration of equation (7) is thus more complicated than that for the subcritical region. Schüle‡ gives a graphical solution which is briefly explained below, and from this the author has deduced a method (also given below) of determining the times of discharge in the supercritical region by calculation alone.

Since

$$p = c \rho^n,$$

we have

$$d \rho = \frac{\rho\, d p}{p\, n}$$

so that equation (7) can be written as

$$C_d A \beta p_1^{\frac{1}{2}} v_1^{\frac{n}{2}} d t = -\frac{V v^{\frac{n-1}{2}}}{p} \frac{d p}{n}$$

Or since

$$p_1 v_1^n = p v^n,$$

$$C_d A \beta p_1^{\frac{1}{2n}} v_1^{\frac{1}{2}} d t = -\frac{V}{n} p^{\frac{1}{2n} - \frac{3}{2}} d p \quad . \quad (9)$$

Also

$$d p = -\frac{p^2}{p_b} d\left(\frac{p_b}{p}\right)$$

so that equation (9) gives the time of discharge as

$$t = \frac{V}{n C_d A p_1^{\frac{1}{2n}} v_1^{\frac{1}{2}}}\int_{\frac{p_b}{p_1}}^{\frac{p_b}{p_2}} \frac{p^{\frac{1}{2n}+\frac{1}{2}}}{p_b\, \beta} d\left(\frac{p_b}{p}\right)$$

$$= \frac{V}{n C_d A \left(\frac{p_1}{p}\right)^{\frac{1}{2n}-\frac{1}{2}} \sqrt{R T_1}}\int_{\frac{p_b}{p}}^{\frac{p_b}{p_1}} \frac{d\left(\frac{p_b}{p}\right)}{\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}\beta} \quad (10)$$

‡ See Schüle's "Technische Thermodynamik," Bd. II, pages 366–395.

Schüle determines graphically the value of this integral as follows:—The values of β for p_b/p, ranging from 0·53 to 1, are calculated from equation (4), i.e.,

$$\beta = \sqrt{\frac{2 \times 32\cdot2 \times 1\cdot4}{0\cdot4}\left[\left(\frac{p_b}{p}\right)^{\frac{2}{1\cdot4}} - \left(\frac{p_b}{p}\right)^{\frac{2\cdot4}{1\cdot4}}\right]}$$

The following table gives the calculated β values:

Table I.—*Values of* β

p_b/p	0·53	0·60	0·65	0·70	0·75	0·80	0·85	0·90	0·95	0·98	1·0
β	3·880	3·836	3·750	3·621	3·432	3·175	2·846	2·390	1·761	1·124	0

The values of $\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}$, which occur in the denominator under the integral in equation (10) are also calculated for various p_b/p values, and are given in Table II:—

Table II.—*Values of* $\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}$.

p_b/p	0·53	0·60	0·65	0·70	0·75	0·80	0·85	0·90	0·95	0·98	1·0
$\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}$	0·5702	0·6363	0·6834	0·7293	0·7754	0·8210	0·8662	0·9111	0·9554	0·9830	1·0

The values of β as given in Table I are now multiplied by the corresponding values of $\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}$ as given in Table II. This gives the values of the denominator under the integral in equation (10), and the reciprocals give the values of the expression under the integral, i.e., the values of

$$\frac{1}{\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}\beta} = k, \text{ say.}$$

These k values are given in Table III opposite.

The areas below the k curve when summed by means of a planimeter give the integral

$$z = \int \frac{d\left(\frac{p_b}{p}\right)}{\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}\beta} \quad . \quad . \quad . \quad (11)$$

The change in z accompanying a change in p_b/p can now be read from the graph, and from this the time taken to effect this change is found from equation (10).

Thus

$$t = \frac{V}{n C_d A \sqrt{R T_1}}\left(\frac{p_1}{p_b}\right)^{\frac{1}{2}-\frac{1}{2n}}(z_2 - z_1) \quad . \quad (12)$$

where

$$z_2 - z_1 = \int_{p_1}^{p} \frac{d\frac{p_b}{p}}{\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}\beta}$$

The curve of z values is plotted in Fig. 1 to a base of p_b/p values.

This process can be repeated for other values of n giving a series of z curves, but these lie within such a narrow field that no serious error is involved in accepting the mean value $n = 1\cdot3$ throughout.

Approximate Time of Discharge in the Supercritical Region

The following modification of Schüle's graphical method is suggested by the author, since it enables the results to be obtained by calculation alone. On referring to Fig. 1 it will be noticed that the z curve is very nearly straight for values of p_b/p ranging from 0·528 to 0·95. The equation of the mean straight line over this region is

$$z = 0\cdot406\, p_b/p - 0\cdot215,$$

so that the approximate equation for the time of discharge when the pressure drops from p_1 to p_2 in the supercritical region becomes

Table III.—*Values of* $k = \frac{1}{\left(\frac{p_b}{p}\right)^{\frac{1}{2n}+\frac{1}{2}}\beta}$

p_b/p	0·53	0·60	0·65	0·70	0·75	0·80	0·85	0·90	0·95	0·98	1·0
k	0·4521	0·4096	0·3901	0·3787	0·3758	0·3836	0·4057	0·4591	0·5977	0·9050	∞

Table IV.—*Values of* z *with* $n = 1\cdot3$

p_b/p	0·528	0·55	0·575	0·60	0·625	0·65	0·675	0·70	0·725	0·75
z	0	0·00985	0·0205	0·0310	0·0413	0·0510	0·0608	0·0702	0·0795	0·0890
p_b/p	0·775	0·80	0·825	0·85	0·875	0·90	0·925	0·95	0·975	1·000
z	0·0984	0·1076	0·1172	0·1276	0·1380	0·1490	0·1608	0·1747	0·1918	0·2300

$$t_2 = \frac{V}{n C_d A \sqrt{R T_1}}\left(\frac{p_1}{p_b}\right)^{\frac{1}{2}-\frac{1}{2n}}(z_2 - z_1)$$

$$= \frac{V}{n C_d A \sqrt{R T_1}}\left(\frac{p_1}{p_b}\right)^{\frac{1}{2}-\frac{1}{2n}}\left\{\left(\frac{0\cdot406\, p_b}{p_2} - 0\cdot215\right) - \left(\frac{0\cdot406\, p_b}{p_1} - 0\cdot215\right)\right\}$$

$$= \frac{0\cdot406\, V\, p_b}{n C_d A p_2 \sqrt{R T_1}}\left(\frac{p_b}{p_1}\right)^{\frac{n+1}{2n}}(p_1 - p_2) \quad . \quad . \quad (13)$$

This equation is applicable if the ratio p_b/p does not exceed 0·95. The case in which p_b/p exceeds 0·95 is considered below.

If the expansion starts at or above the critical pressure, then $p_b/p_1 = 0\cdot528$, so that equation (13) becomes

$$t_2 = \frac{0\cdot406\, V (p_1 - p_2)}{n C_d A p_2 \sqrt{R T_1} \times 1\cdot893^{\frac{n+1}{2n}}} \quad . \quad . \quad . \quad (14)$$

The graph of $1\cdot893^{\frac{n+1}{2n}} \times n$, which occurs in the denominator of this expression, has been plotted to a base of n values in Fig. 2. It is practically a straight line, the equation of which is

$$y = 1\cdot315\, n + 0\cdot578,$$

and hence equation (14) can be written in the simpler form

$$t_2 = \frac{0\cdot406\, V (p_1 - p_2)}{(1\cdot315\, n + 0\cdot578) C_d A p_2 \sqrt{R T_1}} \quad . \quad . \quad (15)$$

Approximate Treatment for Values of p_b/p Lying Between 0·95 and 1·0

For values of p_b/p lying between 0·95 and 1·0 the equation of the mean straight line through the z curve in Fig. 1 is given by

$$z = 0\cdot92\, p_b/p - 0\cdot702,$$

so that in this region the time of discharge when the pressure drops from p_1 to p_2 is given approximately by

$$t_3 = \frac{V}{n C_d A \sqrt{R T_1}}\left(\frac{p_1}{p_b}\right)^{\frac{1}{2}-\frac{1}{2n}}(z_2 - z_1)$$

$$= \frac{V}{n C_d A \sqrt{R T_1}}\left(\frac{p_1}{p_b}\right)^{\frac{1}{2}-\frac{1}{2n}}\left\{\left(0\cdot92\frac{p_b}{p_2} - 0\cdot702\right) - \left(0\cdot92\frac{p_b}{p_1} - 0\cdot702\right)\right\}$$

$$= \frac{0\cdot92\, V}{n C_d A \sqrt{R T_1}}\left(\frac{p_b}{p_1}\right)^{\frac{n+1}{2n}}(p_1 - p_2) \quad . \quad . \quad (16)$$

If the expansion commences with $p_b/p = 0\cdot95$, equation (16) becomes

$$t_3 = \frac{0\cdot92\, V (p_1 - p_2)}{1\cdot053^{\frac{n+1}{2n}}\, n C_d A p_2 \sqrt{R T_1}} \quad . \quad . \quad . \quad (17)$$

The function $1\cdot053^{\frac{n+1}{2n}} \times n = y$, which occurs in the denominator of equation (17), has been plotted to a base of n values in Fig. 3. The equation of the resulting curve, which is very nearly a straight line, is

$$y = 1\cdot02\, n + 0\cdot033.$$

Inserting this in equation (17) gives

$$t_3 = \frac{0\cdot92\, V (p_1 - p_2)}{C_d A p_2 \sqrt{R T_1}\,(1\cdot02\, n + 0\cdot033)} \quad . \quad . \quad (18)$$

The following example serves to show the application of these equations :—

Air, at a pressure of 114·8 lb. per square inch absolute, and a temperature of 297 deg. Cent. absolute, is discharged from a vessel having a capacity of 2·95 cubic feet through a convergent-parallel nozzle having a cross-sectional area of $2\cdot 86\times 10^{-4}$ square feet. It is required to calculate the time taken for the air pressure in the vessel to drop to that of the external pressure of 14·8 lb. per square inch absolute. The coefficient of discharge for the nozzle may be taken equal to 0·98.

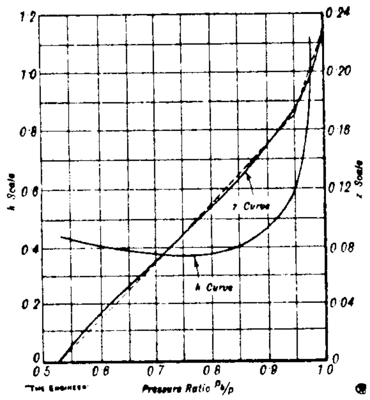


FIG. 1

The time actually taken was measured experimentally and found to be 36 seconds.

The critical pressure, referred to the external pressure, is $p_{cr}=14\cdot 8/0\cdot 528=28\cdot 0$ lb. per square inch absolute. The mean index of expansion for this pressure drop and for the given nozzle was found experimentally to be $n=1\cdot 10$. The time of discharge for this subcritical period is given by equation (8), *i.e.*,

$$t_1=\frac{2\,V}{(n-1)\,C_d\,A\,\beta\,\sqrt{R\,T_1}}\left\{\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{2n}}-1\right\}$$

Here $p_2=p_{cr}=28\cdot 0$ lb. per square inch absolute; $V=2\cdot 95$; $A=2\cdot 86\times 10^{-4}\text{ft.}^2$; $\beta=3\cdot 90$; $T_1=297$ deg. Cent. absolute; $R=96\cdot 3$; $n=1\cdot 10$, $C_d=0\cdot 98$.

These give

$$t=\frac{2\times 2\cdot 95}{0\cdot 10\times 0\cdot 98\times 3\cdot 90\times 2\cdot 86\times 10^{-4}\sqrt{96\cdot 3\times 297}}\left[4\cdot 09^{\frac{0\cdot 10}{2\cdot 20}}-1\right]=21\cdot 1\text{ sec.}$$

The time of discharge in the supercritical period, *i.e.*, while the pressure decreases from 28·0 lb. per square inch absolute to the external pressure of 14·8 lb. per square inch absolute, is given by equation (12), which, with $z_1=0$ and $p_1=p_{cr}$, becomes

$$t_2=\frac{V\,z}{n\,C_d\,A\,\sqrt{R\,T_{cr}}}\left(\frac{p_{cr}}{p_b}\right)^{\frac{1}{2}-\frac{1}{2n}}$$

The value of T_{cr} is found from

$$T_{cr}=T_1\left(\frac{p_{cr}}{p}\right)^{\frac{n-1}{n}}=\frac{297}{4\cdot 09^{0\cdot 10/1\cdot 10}}=262 \text{ deg. Cent.}$$

absolute.

The value of z is found from Fig. 1 for the ratio $p_b/p=\frac{14\cdot 8}{14\cdot 8}=1$, and equals 0·23. The index of expansion n is reduced in this region and is taken as 1·09.

Hence,

$$t_2=\frac{2\cdot 95\times 0\cdot 23}{1\cdot 09\times 0\cdot 98\times 2\cdot 86\times 10^{-4}\sqrt{96\cdot 3\times 262}}\times\left(\frac{28\cdot 0}{14\cdot 8}\right)^{\frac{1}{2}-\frac{1}{2\cdot 18}}=14\cdot 4\text{ sec.}$$

The total time of discharge is thus $t=t_1+t_2=21\cdot 1+14\cdot 4=35\cdot 5$ sec.

The percentage error in the calculated time is thus

$$\left(\frac{36-35\cdot 5}{36}\right)100=\frac{0\cdot 5\times 100}{36}=1\cdot 4\text{ per cent.}$$

In making use of the approximate method it is

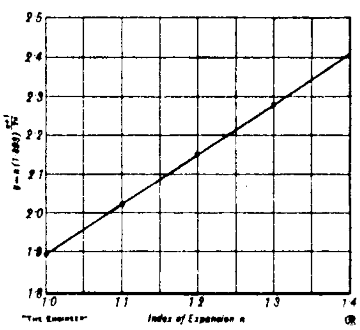


FIG. 2

necessary to consider the expansion in the supercritical region as occurring in two stages. The first stage occurs with a drop in pressure from the critical value of 28 lb. per square inch absolute to the pressure $p=p_b/0\cdot 95=14\cdot 8/0\cdot 95=15\cdot 6$ lb. per square inch absolute. The time for this period is given by equation (15), *i.e.*,

$$t_2=\frac{0\cdot 406\,V\,(p_1-p_2)}{(1\cdot 315\,n+0\cdot 578)\,C_d\,A\,p_2\,\sqrt{R\,T_1}}$$

Here $p_1=p_{cr}=28\cdot 0$ lb. per square inch absolute; $p_2=p=15\cdot 6$ lb. per square inch absolute; $n=1\cdot 09$;

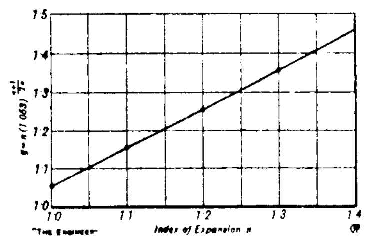


FIG. 3

$T_1=T_{cr}=262$ deg. Cent. absolute. The other values are as given above.

Hence

$$t_2=\frac{0\cdot 406\times 2\cdot 95\,(28\cdot 0-15\cdot 6)}{(1\cdot 315\times 1\cdot 09+0\cdot 578)\,0\cdot 98\times 2\cdot 86}\times 10^{4}\times 15\cdot 6\sqrt{96\cdot 3\times 262}=10\cdot 7\text{ sec.}$$

For the second stage in the supercritical region equation (18) gives

$$t_3=\frac{0\cdot 92\,V\,(p_1-p_2)}{C_d\,A\,p_2\,\sqrt{R\,T_1}\,(1\cdot 02\,n+0\cdot 033)}$$

Here $p_1=15\cdot 6$ lb. per square inch absolute; $p_2=14\cdot 8$ lb. per square inch absolute; $n=1\cdot 09$. T_1 is the temperature at the beginning of this stage and is found from

$$T_1=\frac{T_{cr}}{\left(\frac{p_{cr}}{p_1}\right)^{\frac{n-1}{n}}}=\frac{262}{\left(\frac{28}{15\cdot 6}\right)^{\frac{0\cdot 09}{1\cdot 09}}}=250\text{ deg. Cent.}$$

absolute. These give

$$t_3=\frac{0\cdot 92\times 2\cdot 95\,(15\cdot 6-14\cdot 8)}{0\cdot 98\times 2\cdot 86\times 10^{-4}\times 14\cdot 8\sqrt{96\cdot 3\times 250}\,(1\cdot 02\times 1\cdot 09+0\cdot 033)}=2\cdot 9\text{ sec.}$$

The time of discharge in the supercritical region, as given by the approximate method, is thus $t_2+t_3=10\cdot 7+2\cdot 9=13\cdot 6$ sec., as against 14·4 sec. found by the graphical method. The percentage difference is $\frac{(14\cdot 4-13\cdot 6)}{14\cdot 2}\times 100=\frac{0\cdot 8\times 100}{14\cdot 2}=5\cdot 6$ per cent.

On the total period the percentage difference is $\frac{(35\cdot 5-34\cdot 7)}{35\cdot 5}\times 100=2\cdot 3$ per cent.

ENGINEERING

OCT. 13, 1939 421

LETTERS TO THE EDITOR.

THE KADENACY SYSTEM OF SCAVENGING.

TO THE EDITOR OF ENGINEERING.

SIR,—A recent article in your columns by Dr. G. F. Mucklow (see page 187, *ante*) may give some of your readers an inadequate impression of the scope and nature of the Kadenacy system. I hope you will give me space in your columns to set out very briefly the basic physical principles of the Kadenacy system as I see them.

1. Every sudden change in a physical system is followed by a transient or transition period before the steady state conditions are established. If the exhaust orifice of a two-cycle engine is opened relatively slowly, the steady flow conditions are progressively established, and at any instant the flow of the gases through the valve and along the exhaust pipe can be calculated approximately by the conventional steady-flow theory. The calculations are familiar to all engineers with experience in the design of internal-combustion engines. If the exhaust valve is opened vety rapidly, the transient effects predominate at first and the discharge occurs explosively or ballistically. An entirely different set of physical phenomena accompany this type of discharge. In the Kadenacy system these ballistic phenomena are deliberately established, controlled and utilised to effect the evacuation and re-charging of the cylinder.

2. Each explosive discharge wipes out the aftermath of previous discharges. A Kadenacy engine will, therefore, carry through a single cycle of exhaust and recharge. There is no question of establishing favourable resonances of the exhaust-pipe system, for which a timed series of impulses is essential.

3. The explosive or ballistic discharge takes place as a high-velocity mass flow of the cylinder contents along the exhaust pipe. A truly ballistic discharge forms a steep-fronted compression impulse travelling at a velocity in excess of the velocity of sound in the gas. Theory shows that this velocity increases with increase of the initial pressure ratio across the valve. The phenomena of ballistic discharge are complex. High-frequency waves and vibrations are generally superposed on the mass movement of the gas columns. Their effect on engine performance is quite negligible, and the results obtained by the Kadenacy system can only be interpreted by concentrating attention on the mass movements of the gas. Pressure-time records show up these waves and vibrations very clearly and great care must be taken in the interpretation of such records.

4. The sudden release of a mass of compressed gas sets up a compression impulse travelling outwards with a super-acoustic velocity. The compression impulse must be followed by a rarefaction. This proposition is quite general. Lord Rayleigh (*Dynamical Theory of Sound*, Lamb. Paragraph 71), says, ". . . . a compressed or a rarefied wave cannot exist alone." Professor Lamb, (*Theory of Sound*, vol. 2, paragraph 279) brings out the same point. Professor Sir James B. Henderson, writing on the Kadenacy system, says, "As soon as I became acquainted with the Kadenacy invention, I recognised that he was making use of a well-known phenomenon in explosions, namely, the implosion which immediately follows every explosion, due to the inertia of the explosive gases." There is no need to postulate any mechanism of pipes, etc., to account for the rarefaction which follows the compression impulse.

5. An opposed-piston two-cycle engine, incorporating the Kadenacy inventions, has developed 142 lb. per square inch b.m.e.p., with natural aspiration from the atmosphere, that is, without a blower. The measured induced charge was about 1·5 cylinder volumes of which about 1·15 cylinder volumes were retained in the cylinder. This result cannot be explained on any steady-flow theory.

The general principles outlined above represent a point of view from which the special properties of the Kadenacy engines can be interpreted. If the claims of the inventor, M. Kadenacy, and the actual results achieved with a variety of converted engines are considered from this angle, the Kadenacy system is seen to be logical and consistent.

Yours faithfully,
F. D. SMITH, D.Sc.

Curfew House,
Thames Street,
Windsor, Berks.
October 3, 1939.

ENGINEERING

OCT. 6, 1939 378

PHENOMENA OF THE EXHAUST OF INTERNAL-COMBUSTION ENGINES.

By PROFESSOR SIR JAMES B. HENDERSON, D.Sc., LL.D.

IN engineering practice it is highly desirable to have some theory upon which designers can base their calculations. From time to time these theories should be analysed because, in some cases, the assumptions underlying the theory are forgotten and the theory is applied to problems in which these assumptions do not hold. Such a case has arisen in the theoretical treatment of the exhaust of an internal-combustion engine, as given in text-books on the subject. There is no branch of engineering science in which theory lags farther behind practice than in problems involving the dynamics of fluid motion. The mathematical science of hydrodynamics, when applied to such a practical problem as the resistance to the motion of a ship through water and the power required to overcome this resistance, shows that the resistance can be divided into that absorbed by wave motion of two kinds, and that absorbed by skin-friction; but the practical science contains the results of many thousands of experiments plotted in curves, and considerable experience is required in dealing with these.

In aerodynamics the mathematical science lags even farther behind the practical science than in hydrodynamics, because of the compressibility of the medium. Water may be considered in most practical problems as an incompressible, non-viscous fluid, like the ideal fluid assumed in mathematical theory, but air is very compressible and has thirteen times the viscosity of water relatively to its density, so that compressibility and viscosity cannot be ignored in aerodynamic problems and the mathematical problems, therefore, become more difficult; hence the greater lag between theory and practice in this branch. One has only to examine the motion of the surface of the sea on a calm day from the deck of a ship under way, the separate sets of waves and the very turbulent motion astern, to realise that theory cannot deal with the whole problem and can only act as a guide. In the motion of aircraft the phenomena are still more complicated from the mathematical point of view, and for high velocities, such as arise in the motion of projectiles, the science is mostly empirical. The same applies to phenomena of the nature of explosions in the air. It is possible to calculate the maximum pressure due to a detonation of explosive placed, say, against an armour plate, but the resulting motion produced in the air is beyond the powers of mathematical science.

One gets some idea of the complicated motion in the exhaust of an engine by close observation of the swirling motion of smoke issuing from a chimney, or the more complicated motion of the steam issuing from the funnel of a locomotive, or the still more complicated motion of the smoke cloud issuing from a big gun after the projectile has left the muzzle, remembering that, although a local cloud of smoke is all that is obvious, part of the energy of the exhaust may be smashing windows miles away. Text-book theory ignores what is happening outside the exhaust valve of the engine, except that it assumes a constant back pressure in the exhaust pipe, and assumes that the equation which is used for the flow of gas or steam through a nozzle, from a vessel in which

the pressure is kept constant, applies at every instant during the exhaust. This formula is based upon the assumption of a steady state in the vessel, so that the distribution of energy in the vessel may be ignored and the kinetic energy in the issuing jet equated to the work done in maintaining the constant pressure in the vessel. For all practical purposes this formula gives results agreeing with experiment within a few per cent., when applied to the conditions of a steady state as defined above. A discussion of the discrepancies between theory and experiment in the steady state is to be found in the *Proceedings* of the Mechanical Engineers for 1913, in a paper written by the present writer at the request of the Council of that Institution.

The exhaust of an engine is a transient phenomenon; it is all over in a few thousandths of a second. Hence a great deal of justification is required in applying to it a formula relating to a steady state, in which it is assumed that the energy in any particular unit of the gas remains in the same unit throughout its passage through the vessel and nozzle, and that no unit gains energy from or loses it to neighbouring units; in other words, that all progressive wave phenomena in the vessel are excluded. The only justification which can be given is that there is no other theoretical basis for a solution. It is surely far better to recognise the fact that theory cannot help us, and to deal with the problem empirically as is done in other branches of engineering. We shall see later that the application of the above theory has only hampered progress.

If we take an inflated child's balloon and puncture it with a pin, the balloon will slowly deflate,* and the pressure inside will gradually fall to atmospheric. If the hole is large, the balloon explodes with a loud noise and collapses completely. The gas in exploding leaves a partial vacuum behind, due to its inertia, and repeated condensations and rarefaction follow in the balloon. If we break an incandescent electric lamp of the vacuum type, there is an implosion first of all, then an explosion followed by repeated condensations and rarefactions creating a noise. When a gun is fired, the explosion of the gases out of the muzzle behind the projectile leaves a partial vacuum in the gun behind it, causing air to rush in; but in all these transient phenomena it must be remembered that the time interval between explosion and the resulting implosion, or vice versa, is so short that the eye cannot follow it, and special methods have to be devised to demonstrate them. These may be dealt with in a subsequent article.

The question naturally arises: Is there no way in which theory can help in dealing with the problem of exhaust of an engine? Theory indicates first of all that, in a transient phenomenon like the sudden explosion from the exhaust valve of an engine, the acceleration phenomena in the gas both inside and outside the cylinder, in other words the wave motion, are of paramount importance. The moment the valve is opened continuity is established between the gas inside and outside the cylinder, and we cannot deal with one without the other. Mathematics cannot help us at present. There is one way, which occurred to the writer some time ago, in which a problem in hydrodynamics that has been solved and, from the energy point of view, is an exact analogy of our problem, may throw considerable light upon it. I refer to the generation of a wave of translation in a canal by the sudden emptying of a lock. The still water behind the lock gates is the counterpart of the gas in the cylinder. Every unit of water in the lock possesses the same energy, due to height or pressure, just as every unit of gas in the cylinder possesses the same energy due to pressure and temperature. In the laboratory experiment to show the wave of translation, the gate of the lock can be raised, thereby opening a sluice the whole width of the canal. If the sluice is raised very little the lock empties very slowly and Bernoulli's equation can be applied to the rate of flow, the surface of the water in the lock remaining practically horizontal.

If the sluice is raised quickly to near the external surface in the canal, the phenomena change completely. The water flows quickly through the sluice, raising the external level and lowering the internal one, but the internal surface level falls faster near the sluice end than at the upper end, forming a wave surface in the lock. The surfaces both in and outside the lock are disturbed by eddies and small waves. The long wave in the lock soon reverses, the surface level becoming lower at the upper end; the final discharge may expose the bottom of the lock and a counterflow into the lock from the canal supervenes. The chief phenomena in the canal some distance from the lock are the existence of a wave of translation moving along it, which may be followed by a smaller negative wave of the same type moving at a lower speed, the two waves separating slowly.

A wave of translation only occurs in shallow water and differs from a sea wave in that, in the former, the whole of the energy is transmitted, whereas in the latter only half is transmitted, and half left behind. The energy at any section of the wave is measured by the surface level at all times, both when in the lock and in the canal. The slow-moving phenomena in the canal give us a clear conception of the invisible phenomena which take place both in the cylinder and exhaust pipe of an engine when the exhaust valve is opened, and are all over in a few thousandths of a second.

My attention was drawn to this subject by seeing the experiments with the Kadenacy engine. Kadenacy makes use, for the first time, in a two-stroke cycle engine, of the rarefaction in the cylinder following the explosive discharge of the exhaust to suck in the fresh charge, and succeeds in supercharging to such an extent that he can eliminate the charging compressor and obtain a larger horsepower than was obtained with it.

* This is not our experience.—Ed., E.

ENGINEERING

JAN. 5, 1940

SUDDEN DISCHARGE OF AIR FROM A PRESSURE VESSEL.

By S. J. DAVIES, D.Sc.

THE object of the experiments described here was to study the pressure conditions in a vessel following the sudden discharge of compressed air from the vessel. The apparatus used in the experiments is illustrated in Figs. 1, 2 and 3, page 18. It consists of a cylinder having an internal diameter of 4 in. and length of approximately 13 in. The cylinder is closed at one end, and is provided at the other end with an orifice $3\frac{5}{8}$ in. diameter, in which is seated a valve having a masking $\frac{3}{4}$ in. long. For convenience, a cylinder used in previous experiments was employed. It will be noticed from Fig. 3 that the closed end contains a shallow recess, and that the valve embodies a cylindrical internal extension. These modifications did not in any way affect the validity of the experimental results. The external mechanism shown on the left in the figures is provided for holding the valve on its seating while the vessel is being charged with air to the required pressure, and for releasing the valve when desired. The valve is forced from its seating by the pressure of the air in the vessel and thus uncovers the orifice, the masking ensuring that the valve will have attained a high speed before the end of the masking clears the discharge orifice. This valve mechanism, as will be seen in Fig. 3, comprises a spring-loaded lever which presses on the valve in the closed position and can be disengaged from the valve by striking an external tripping arm. The photographs reproduced in Figs. 1 and 2 show the valve in the closed and open positions, respectively.

The experiments consisted in charging the vessel with air at varying pressures, releasing the valve, and making observations of the minimum pressures reached in the vessel at the points marked A, B, C and D, in Fig. 3, and of the time intervals. These observations were made by means of the "Sunbury" cathode-ray oscillograph, made by Messrs. The Standard Telephone Company. To obtain the greatest possible accuracy, the calibrating unit of this instrument was used for measuring the minimum pressures reached in the vessel. For this purpose, provision was made to place the pressure elements and the calibrating unit, respectively, at the points A, B, C and D, and contacts were arranged so that the horizontal movement of the spot across the screen of the oscillograph commenced at the instant the end of the valve masking cleared the discharge orifice of the vessel.

As a first experiment, the vessel was charged with air to a series of initial pressures ranging from 0 lb. per square inch gauge to 60 lb. per square inch gauge, and the valve then released. In each case the value of the minimum pressure reached at the point A was measured by means of the calibrating unit of the indicator. The values thus obtained are recorded in the form of a curve of minimum pressure against initial pressure in Fig. 4, and it will be seen that, for all initial pressures above 1 lb. per square inch gauge, a depression was recorded in the vessel at A, following the discharge of the compressed air, the depression for the initial pressure of 60 lb. per square inch gauge reaching the value of 4·3 lb. per square inch.

In a second experiment, the vessel was charged with air to an initial pressure of 40 lb. per square inch gauge and, following the release of the valve and the discharge of the compressed air, the value of the depression reached at the points B, C and D (in addition to A) were measured. The values thus obtained are shown in Fig. 5 as ordinates on a base represented by the vessel itself, and it will be seen that the depression is greatest at the closed end of the vessel and least at the discharge orifice.

In order to obtain an indication of the time of occurrence of the maximum depression at points A, B, C and D in the second experiment, the calibrator diagram was photographed at each point with an applied static depression slightly less than the maximum depression occurring at the point concerned. It was found that at each point in the vessel the maximum depression was reached about 0·006 second after the commencement of opening of the orifice, but the accuracy of the time measurement was not sufficient to permit conclusions to be drawn concerning the exact instants of time at which the maximum depressions occur relative one to the other.

By repeating this experiment with atmospheric pressure applied to the calibrating unit, it was found that at all points the pressure in the vessel fell below atmospheric pressure about 0·0035 second after the commencement of opening of the orifice, and returned again to atmospheric pressure about 0·007 second after the commencement of valve opening.

The apparatus is designed to give a rapid opening of a large area of discharge orifice in the vessel. In order to obtain a measurement of the time taken to open the orifice fully in the second experiment, two time points were determined, one when the valve masking commences to clear the orifice and the other at the full opening of the orifice, *i.e.*, when the end of the valve masking had moved $\frac{23}{32}$ in. away from the orifice. This time interval was found to be 0·0019 second, showing that the orifice is already fully open before a depression is reached at any point in the vessel. By way of interest, it is indicated that the time for full opening at 20 lb. initial pressure was 0·0026 second, and at 60 lb. initial pressure 0·0015 second.

Conclusions.—The experiments show that the sudden discharge of compressed air from the vessel leaves a depression in the vessel even with the lowest initial pressures, and that this depression increases as the initial pressure increases, (experiment 1.) The distribution of pressure in the vessel shows that the depression is greatest at points most remote from the orifice, (experiment 2). The time intervals in which the depression occurs are short; the first fall below atmospheric pressure begins some 0·0035 second after the commencement of opening the orifice and the existence of the depression is not of long duration; certainly no longer than 0·0035 second.

DISCHARGE OF AIR FROM A PRESSURE VESSEL

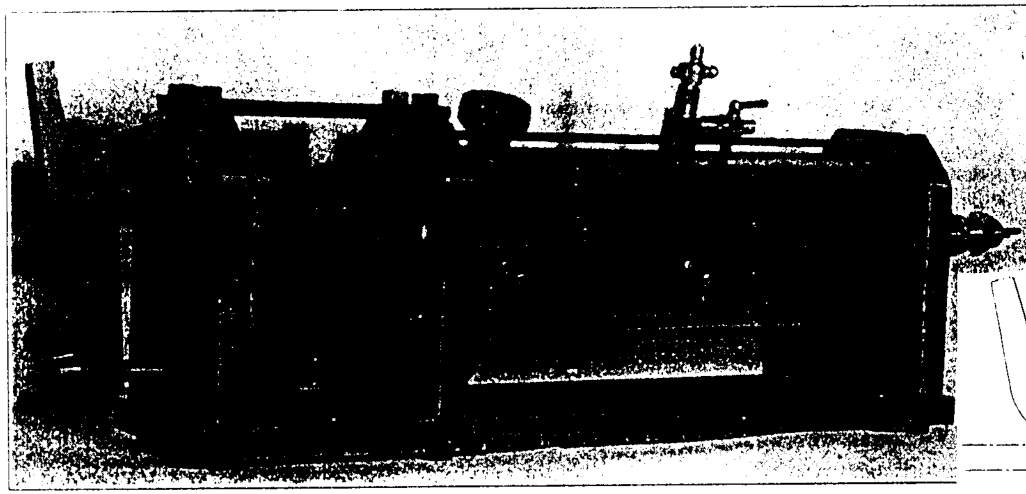


Fig. 1. Cylinder with Valve Closed.

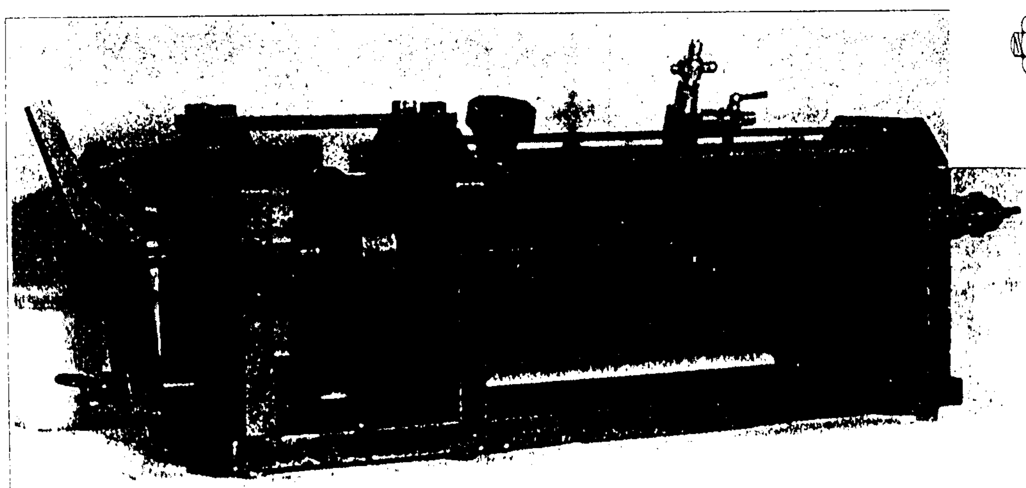
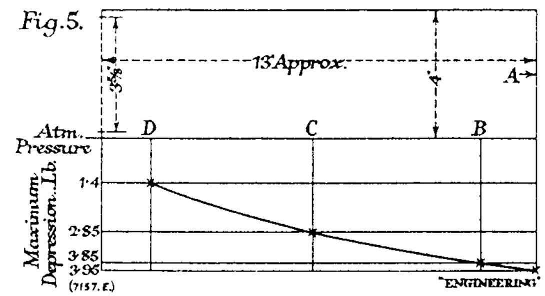
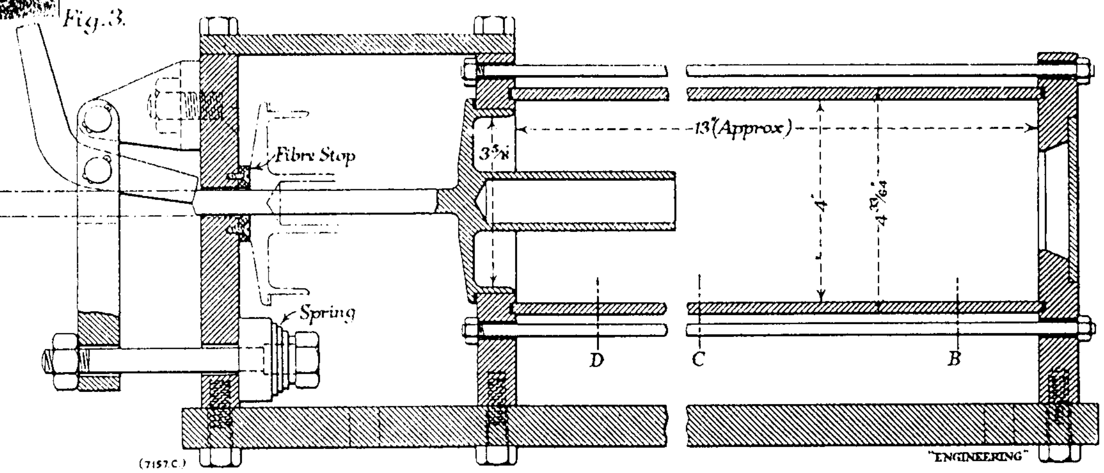
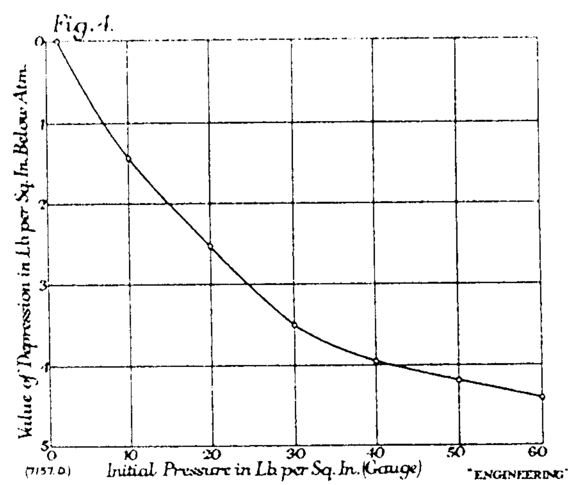


Fig. 2. Cylinder with Valve Open.







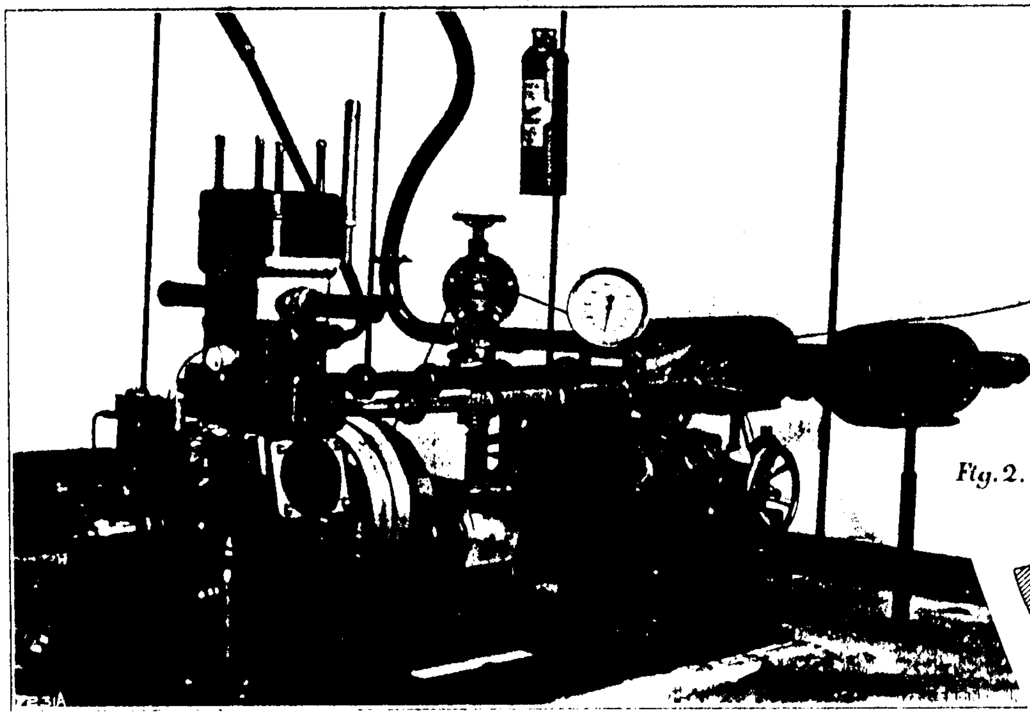


FIG. 1. EXPERIMENTAL ENGINE.

ENGINEERING

MAY 24, 1940 515

AN ANALYSIS OF CERTAIN CHARACTERISTICS OF A KADENACY ENGINE.

By PROFESSOR S. J. DAVIES, D.Sc., M.I.Mech.E.

THE present report is a continuation of one* prepared by the author about three years ago on vo-stroke engines of Kadenacy design, which, will be remembered, are being developed by essrs. Armstrong-Whitworth Securities Company, Limited, Thames House, London, S.W.1. In that report, the author described various tests he had made to illustrate the characteristics of Kadenacy engines, and the experimental methods were carried to such a point that hand-drawn diagrams from a cathode-ray apparatus were included to show the variations in static pressure at various points in the exhaust system. Since those tests were made, it has been possible to improve the measuring technique very considerably, and, in the present report, certain of the characteristics of a Kadenacy engine are analysed in detail.

In this investigation, it is desirable to deal objectively with the motion of the air for combustion to the engine cylinder, and with the evacuation of the exhaust gases from the cylinder later in the cycle; in other words, to consider the general movement of gases through the engine. The engine under test was an opposed-piston two-stroke oil engine, of 65 mm. bore and 210 mm. combined stroke, the lower piston working directly through a connecting rod on the crankshaft and controlling the exhaust ports, while the upper piston was connected to the crankshaft by return connecting rods and controlled the admission ports. A photograph of the engine is reproduced in Fig. 1, above, from which it is seen that the admission passages are at the top, leading directly from the atmosphere, and the exhaust system at the bottom. In addition to the readings usually obtained during engine tests, special observations were made of the pressure variations in the admission passages, in the cylinder and in the exhaust system. In making the pressure

Fig. 2.

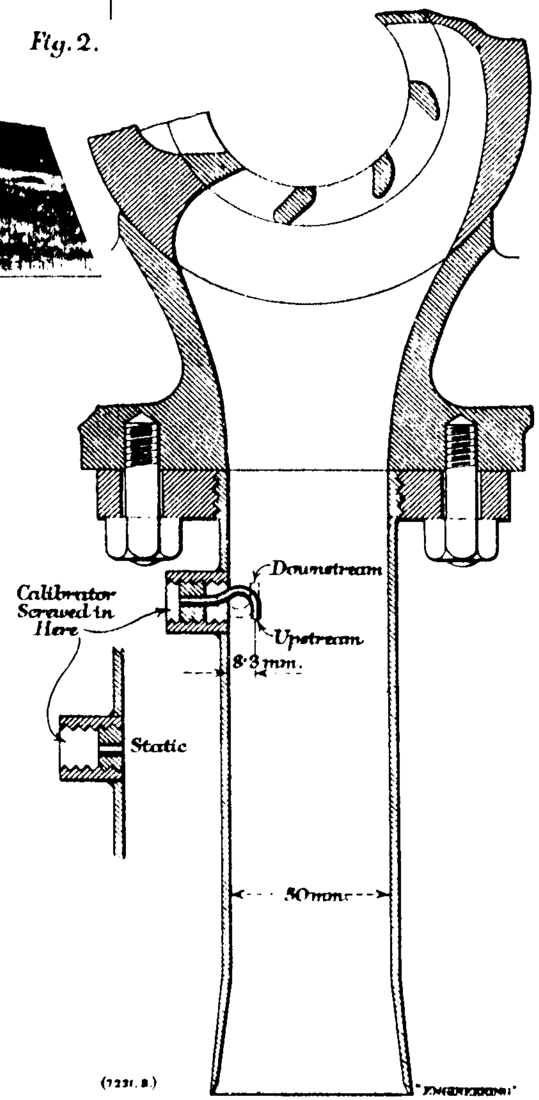
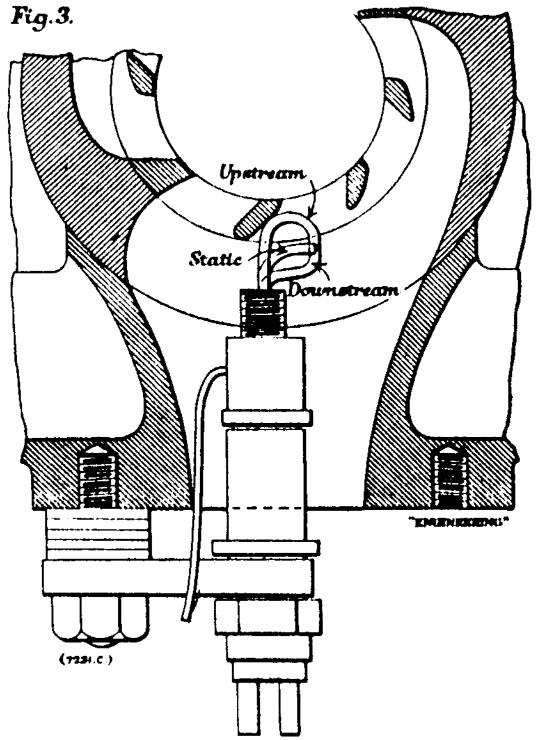


Fig. 3.



* ENGINEERING, vol. 143, page 685, et seq. (1937).

CHARACTERISTICS OF A KADENACY ENGINE.

Fig.4.

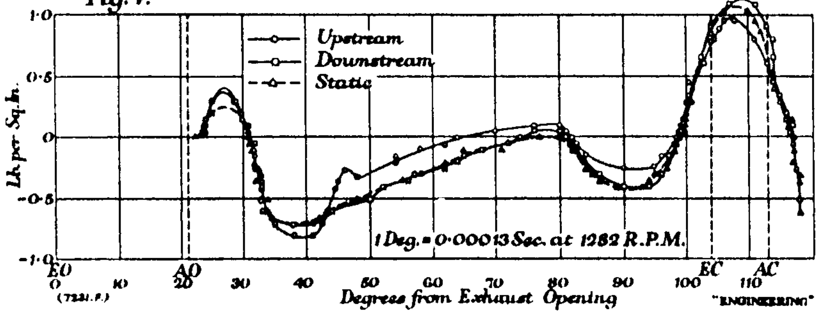


Fig. 6a.

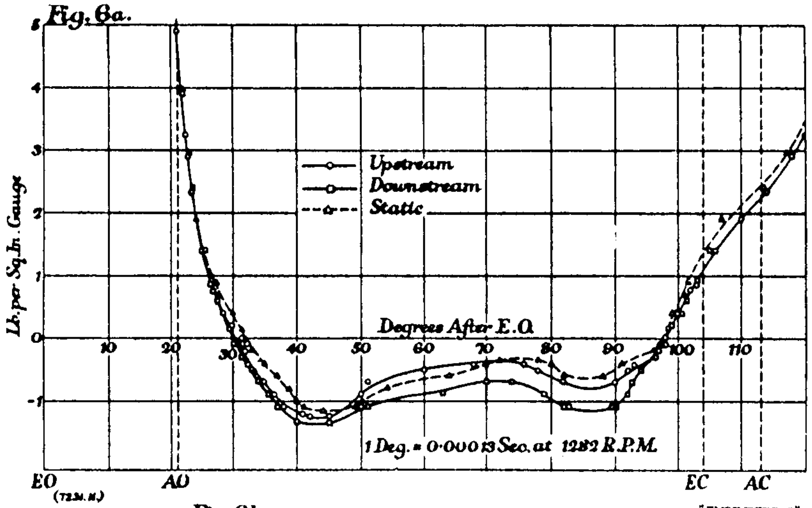


Fig.6b.

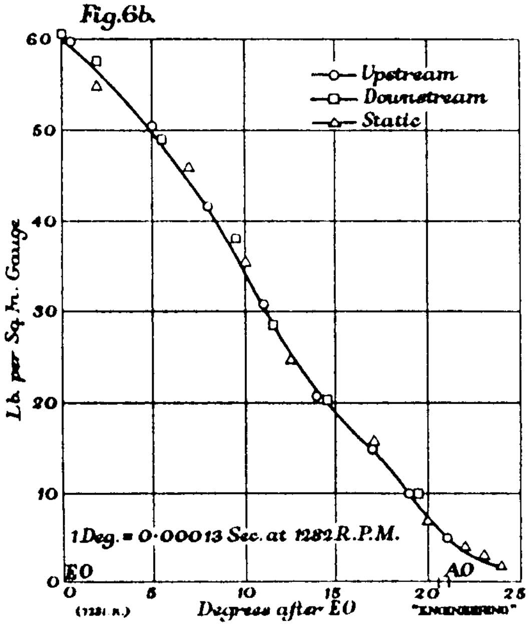


Fig.8.

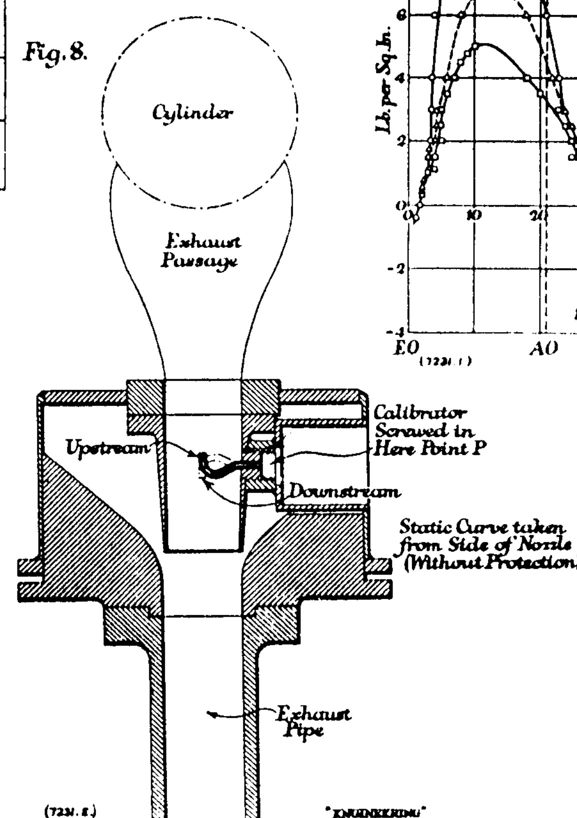


Fig.5.

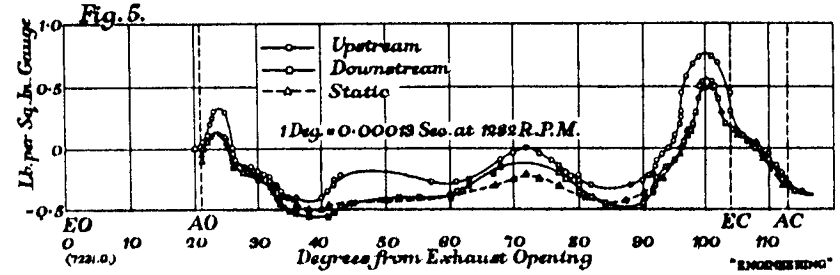


Fig.9.

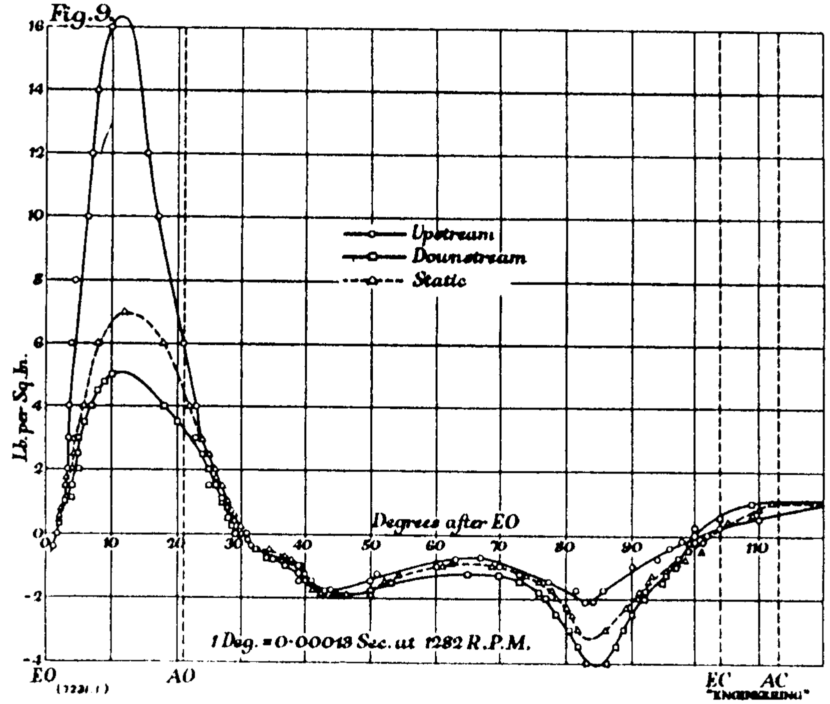
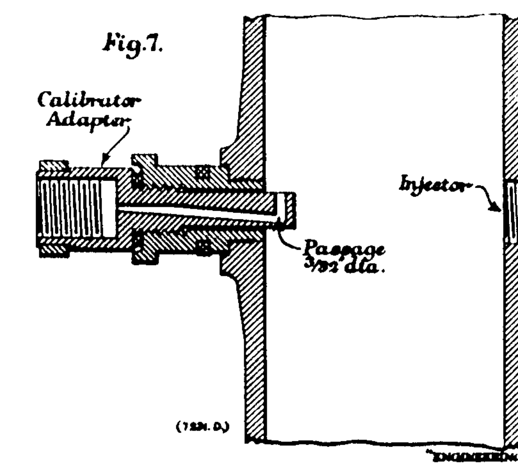


Fig.7.



AN ANALYSIS OF CERTAIN CHARACTERISTICS OF A KADENACY ENGINE.

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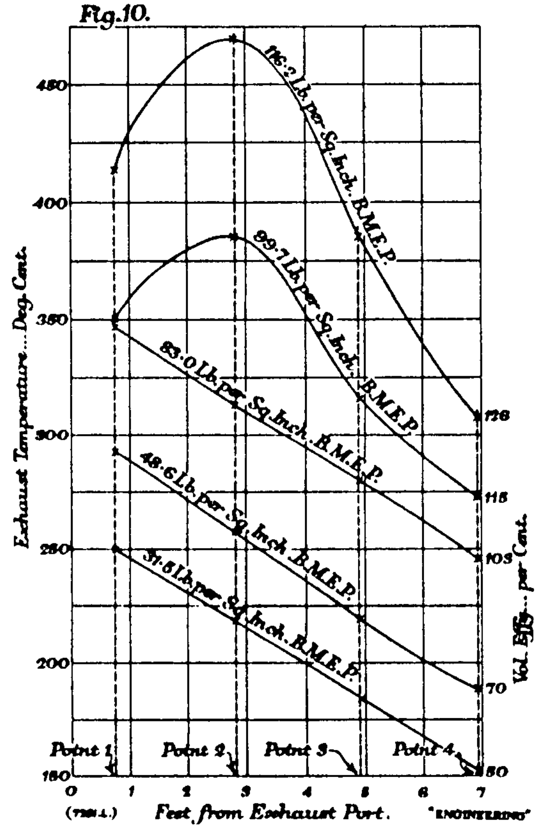
(*Continued from page* 517.)

A FURTHER factor to be taken into account is that, of the air charge taken in, a portion may pass through to the exhaust without taking part in the combustion. This is to be expected from the high value of the volumetric efficiency measured, but other evidence for this may be found in Fig. 10, below, which shows the mean temperatures observed by placing a mercury thermometer in the exhaust pipe at four measuring points, 1, 2, 3 and 4, at distances, respectively, of 9 in., 34 in., 59 in. and 83 in. from the exhaust port. Five curves are shown on this figure, all taken at 1,282 r.p.m., and with the values of B.M.E.P. also shown. The measured values of the volumetric efficiency at these values of B.M.E.P. are given on the right. It cannot be contended that these temperatures are accurate mean values, since the fluctuations of temperature at these points during a cycle take place extremely rapidly, and are very irregular in character. Nevertheless, as comparative values they are useful. In the lowest three curves there is a consistent rise of temperature at measuring point 1, nearest the cylinder, with increase of load, and increase in volumetric efficiency, but in the two upper curves there is inconsistency in the readings at this point, the temperatures shown being much lower than would be expected from the trends of the respective curves. The lower values can only be attributed to the effect of cold air passing through the cylinder and out of the exhaust port to the measuring point during the charging period, this cold air having but a small direct effect at the next measuring point 2, and a still smaller effect beyond 2. That cold air passes into the exhaust pipe in the test forming the subject of this analysis is, therefore, clear, and it remains to be seen how much air is finally retained in the cylinder.

Fig. 10.



A check on the relationship of the temperatures in the cylinder at A C and E O on Figs. 6*a* and 6*b*, page 516, *ante*, may be made as follows: At A C, the contents of the cylinder consist of the air that remains in the cylinder; at E O the exhaust gases forming the cylinder contents consist of this air together with the fuel charge. The volumes enclosed between the pistons at E O and A C are 583 c.c. and 600 c.c., respectively. At E O, the absolute pressure in the cylinder is 74·7 lb. per square inch, while at A C the pressure is 17·2 lb. per square inch. Assuming that the air to fuel ratio for combustion is 23 to 1, which is reasonable in view of the B.M.E.P. of 116·2 lb. per square inch; that the physical properties of the resulting cylinder contents at E O lead to a value of R, the gas constant, equal to 92·2 ft.-lb. per lb. per deg. C.; and that the cylinder contents at A C consist of pure air, giving R equal to 96·0, it is possible to obtain a rough relationship between the absolute temperatures at these points. Let T_{EO} and T_{AC} be the absolute temperatures at E O and A C, respectively. Then at E O,

$$T_{EO} \propto \frac{P \cdot V}{(23+1) \cdot R} = \text{constant} \cdot \frac{74{\cdot}7 \cdot 583}{24 \cdot 92{\cdot}2};$$

and at A C,

$$T_{AC} = \text{constant} \cdot \frac{17{\cdot}2 \cdot 600}{23 \cdot 96{\cdot}0}.$$

$$\frac{T_{EO}}{T_{AC}} = \frac{74{\cdot}7}{17{\cdot}2} \cdot \frac{23}{24} \cdot \frac{96{\cdot}0}{92{\cdot}2} \cdot \frac{583}{600} = 4{\cdot}21,$$

that is, the temperature at E O is rather more than four times that at A C. This relationship depends only slightly on the assumed air to fuel ratio.

In order to arrive at a reasonable figure to assume for the temperature of the cylinder contents at E O, consideration must be given to three points. Firstly, from the temperature and the indicated pressure, the density of the charge can be calculated, and from this the mass of gas enclosed between the pistons may be found. The difference between this mass of gas and the total charge of air per cycle, 0·00235 lb., gives the proportion of the air charge that must have passed through the cylinder into the exhaust system. The proportion left, which receives the full charge of fuel, must be sufficient to give the measured B.M.E.P. Secondly, the air to fuel ratio of the charge must be reasonable. Thirdly, since the temperature at A C is 1/4·21 of that at E O, the value chosen for E O must lead to a suitable figure at A C. Taking all these conditions into account, it was decided to take 1,300 deg. C. abs. as the temperature at E O.

This temperature gives the mass of air and fuel enclosed at E O as 0·001849 lb., which means that a proportion of 75·4 per cent. of the total air charge is retained or trapped in the cylinder; this, in its turn, gives a "trapped" volumetric efficiency of 95·0 per cent. In view of the results generally observed in four-stroke engines, where a B.M.E.P. of 100 lb. per square inch is associated with volumetric efficiencies of the order of 90 per cent., this figure is reasonable for the B.M.E.P. of 116·2 lb. per square inch measured. The corresponding air to fuel ratio is 75·4 per cent. of 30 to 1, or about 23 : 1, agreeing with the ratio assumed above. Lastly, the corresponding temperature at A C is 309 deg. C. abs., a rise of 16 deg. C. from the atmospheric value of 20 deg. C. This is a small rise, but, at 1,282 r.p.m., is consistent with the complete evacuation of the exhaust gases necessary to give the volumetric efficiency of 126 per cent. measured, and the trapping of a pure charge of atmospheric air at A C. It is a low value for the temperature at the beginning of compression, and will lead to a corresponding reduction in the temperature throughout the cycle, which is an advantage in view of the high B.M.E.P. Altogether, the evidence in support of the assumed figure of 1,300 deg. C. abs. is satisfactory.

Having reached this point, we are faced with the most important question: on what basis is the transformation of the energy of the gases in the cylinder into the kinetic energy of the gases passing through the exhaust port to be considered? Reference to books on internal-combustion engines shows the most advanced treatment to be that in *Dieselmaschinen*, page 210, by the late Professor Magg. In this treatment, based on the principles of steady flow from a reservoir, as given by Schüle (*Technical Thermodynamics*, page 215), the exhaust from the cylinder is considered as flow from a vessel through an orifice of varying time-area. The flow is further assumed to take place with a fall of pressure from the cylinder pressure to the port pressure, and from the port to the back pressure, which is atmospheric pressure, the port pressure being that at the smallest cross section of the stream—that is, the "throat" pressure. So long as the ratio of the back pressure to the cylinder pressure is less than the critical value, the gases are considered to flow through the port with the

speed of sound in the medium under the conditions prevailing at the port. Below the critical ratio, the rate of mass discharge depends only on the speed of sound and on the effective area of the port or throat. When the pressure ratio is greater than the critical value, the outflow is less, and then depends upon the pressure difference. Throughout the discharge, the expansion of the gases is assumed to be continually resisted, and the conditions throughout are considered to be strictly adiabatic or isentropic. In effect, Magg made a separate calculation for each of a large number of values of diminishing cylinder pressure and increasing port area, assuming a constant back pressure equal to 1·05 (atmospheric pressure), and he then integrated these over the exhaust period. The condition of the gas at any instant, and at any point in the discharge, was assumed to be the same as would be the case in steady flow from a region of high pressure to one at constant low pressure through a suitably shaped nozzle, as is assumed, for example, in an impulse steam turbine.

Upon examination, it is seen that this treatment is incomplete in connection with the present analysis, since it does not take into account the following circumstances: the rapidity of the changes in the area of the cross section of the port; the rapid changes in the pressure in the cylinder; the relatively small volume of the cylinder compared to the area of the port; the motion of the gases in the cylinder; and the motion and pressures of the gases external to the port. Given a very large volume in the cylinder relatively to the port, and a very slow rate of opening of the port, then the conditions approach those of continuous flow, such as would take place from a reservoir of infinite volume through a port of finite area. At the other extreme, if the port opens instantaneously to an area large in relation to the volume of the cylinder, then, as soon as the port is opened, but before motion of any kind can begin, there is discontinuity between the cylinder pressure and the external pressure. The actual conditions in the engine under test probably lie somewhere between these two extremes, as regards the rate of opening of the port, and the relation of the area of the port to the volume of the cylinder.

Preliminary calculations of the momentum passing point P, Fig. 8, page 516, *ante*, showed that the velocities through the ports are very high, at least during the first 10 deg. after E O. For these high velocities through the ports, there must be high velocities of approach in the cylinder, corresponding to the displacement of the mass of gas involved, and these velocities of approach will exert an influence on the exhaust process, whereas Magg's treatment assumes the general mass of gas in the cylinder to be at rest. The treatment further assumes that the gases passing out of the port can expand freely beyond the port, and pass away into the atmosphere. This is not true for, however short the exhaust pipe, the gases leaving the port possess direction as well as velocity, and, during the exhaust process, must impinge upon the external gases and set them in motion. This action involves some increase of back pressure. In an engine, an exhaust pipe of some length must be provided in order to convey the gases away. That the resistance due to the back pressure can soon become appreciable is shown by the pressures recorded at P, and shown in Fig. 9, page 516, *ante*. Point P is only some 4 in. from the exhaust ports and the area of cross section of the pipe at P is, up to 13 deg. after E O, much more than twice the area of the port, so that the gases should, according to the treatment, continue to expand from the port to point P. The values of the pressure recorded at P show that such a free expansion of the gases from the port outwards cannot be assumed in this case.

It must be concluded that, during the exhaust process, and more especially during the first 21 deg. after E O, the pressures, temperatures, specific volumes, and velocities of the gases are all changing rapidly from instant to instant, and the conditions controlling the inter-conversion of heat energy and mechanical energy in the exhaust gases, on the one hand, and the interchange of momentum between these gases and the external gases, on the other hand, are much more involved than those in Professor Magg's treatment. The author has sought other treatments which would take account of the rapid variations in the conditions, but without success, and is compelled therefore to confine himself to his own experimental data. Using these data, and considering, from instant to instant, the energy still remaining in the cylinder and that conveyed away by the gases issuing from the exhaust port, he has calculated the pressures, temperatures, and velocities necessary for continued equilibrium between the energy in the cylinder and that lost externally.

Taking 1,300 deg. C. abs. as the temperature of the cylinder contents at E O, and 0·001849 lb. as the corresponding mass of these gases, it is possible, by considering the quantities of energy lost from the cylinder and those quantities of energy remaining in the cylinder, and by forming equations from these, which may be solved by step-by-step calculations, to analyse the movements of the gases during the period from E O onwards. In particular, the interval from E O up to A C, and perhaps a little beyond, is of importance, since, during this interval, the passage of the gases out of the cylinder through the exhaust port is not influenced by the entry of gases through the admission port. The variables to be considered at any instant are: (1) the absolute pressure in the cylinder, taken from the indicator diagram; (2) the absolute temperature of the gases in the cylinder; (3) the density of the gases, calculated from (1) and (2); (4) the volume enclosed between the pistons; and (5) the mean effective area through the exhaust port during a given step. The basis of such an analysis is the expansion curve in Fig. 6*b*, given as pressure on a base of time by means of the Sunbury indicator. In view of the special precautions observed in producing the diagram, it may be regarded as having an accuracy of an exceptionally high order.

The form of the indicator diagram during the interval from E O to A O is the result of: (1) the heat transformed into work on the piston, resulting from the pressure exerted, throughout the increase in volume enclosed between the pistons; (2) the heat lost by the gases through the walls and piston crowns; (3) the loss of mass through the exhaust port, and the potential and kinetic energy associated with this mass; (4) the work done by the displacement by these gases of the external gases outside the port; and (5) the kinetic energy existing instantaneously in the gases in the culinder consequent upon the outward movement of the gases through the exhaust port. Taking 1 deg. as a convenient increment, these various quantities can be calculated on a step-by-step basis.

At E O, the gases in the cylinder may, for the purpose of this analysis, be assumed to be at rest. At the beginning of any of the steps after E O, the gases in the cylinder will possess energy due to their temperature, and also kinetic energy. During the step, energy will be lost externally from the cylinder: (1) in work on the pistons; (2) as heat to the walls and the piston; (3) as pressure and potential energy in the mass which has passed out through the exhaust port; and (4) as kinetic energy in this mass. Other changes of the energy in the cylinder may follow from an increase or decrease in the kinetic energy of the gases remaining in the cylinder. The total energy in the gases in the cylinder at the end of a step is equal to the total energy in the gases in the cylinder at the beginning of the step, minus the energy which has been lost externally during the step.

The total energy of a unit mass of gas is represented by

$$\frac{P\,V}{J} + C_v\,.\,T + \frac{1}{J}\,.\,\frac{v^2}{2\,g},$$

where

P = pressure,
V = specific volume,
T = absolute temperature,
v = mean linear velocity, and
C_v = specific heat at constant volume.

Of this energy, however, only the second and third terms may be utilised in expansion from a closed vessel. The general expression may be re-written

as

$$\frac{\mathrm{R\,T}}{\mathrm{J}} + \mathrm{C_v \, . \, T} + \frac{1}{\mathrm{J}} \cdot \frac{v^2}{2\,g},$$

or

$$\mathrm{C}_p \, . \, \mathrm{T} + \frac{1}{\mathrm{J}} \cdot \frac{v^2}{2\,g},$$

where R is the gas constant, and C_p is the specific heat at constant pressure. It is now possible to produce the energy equation in the following manner: Let W_1, W_2 represent the masses of gas in the cylinder at the beginning and end of the step, respectively, and T_1, T_2 the corresponding values of the absolute temperature; v_1, v_2, the mean velocities in the mass of gas in the cylinder; t_1, t_2, the absolute temperatures at the port at the beginning and end of the step; v, the mean velocity of the gases passing through the exhaust port during the step; and H, the heat lost to the walls and pistons during the step. The equation is then:

$$\overset{\text{I.}}{\mathrm{W_1 C_v T_1}} + \overset{\text{II.}}{\frac{4}{3} \cdot \frac{1}{2g} \cdot \mathrm{W_1}\, v_1^2 \frac{1}{\mathrm{J}}} = \overset{\text{III.}}{\mathrm{W_2 C_v T_2}}$$

$$+ \overset{\text{IV.}}{\frac{4}{3} \cdot \frac{1}{2\,g} \cdot \mathrm{W_2}\, v_2^2 \frac{1}{\mathrm{J}}} + \overset{\text{V.}}{(\mathrm{W_1 - W_2})\, \mathrm{C}_p \left(\frac{t_1 + t_2}{2}\right)}$$

$$+ \overset{\text{VI.}}{\frac{\mathrm{W_1 - W_2}}{2\,g} \, . \, v^2 \, . \, \frac{1}{\mathrm{J}}} + \overset{\text{VII.}}{\int \mathrm{P}\, d\,\mathrm{V} \, . \, \frac{1}{\mathrm{J}}} + \overset{\text{VIII.}}{\mathrm{H}}.$$

For convenience, each term in the equation is marked by a Roman numeral above it.

Terms I and III of this equation represent the potential energy of the gases in the cylinder at the beginning and end of each step. Terms II and IV represent the kinetic energy of the mass of gas in the cylinder, where v_1, v_2 are the mean velocities of the centre of mass, the distribution of velocity being taken as linear from 0 to 2 v_1 and to 2 v_2, respectively. Term V represents the energy (potential energy + work done) of the gas $(W_1 - W_2)$ escaping from the cylinder during the step. Term VI represents the kinetic energy of the gas $(W_1 - W_2)$ escaping from the cylinder during the step with a mean velocity v through the port. Term VII represents the work done by the gas in the cylinder on the pistons during the step. Term VIII represents the heat lost by the gases in the cylinder to the walls during the step, and is estimated to be one-quarter of the work done during the step, as obtained from the indicator diagram, so that terms (VII + VIII) $= \frac{5}{4} \cdot$ (work done). It will be found convenient for the solutions to transpose terms III, VII and VIII, and the equation may now be re-written:

$$\overset{\text{I.}}{\mathrm{W_1 C_v T_1}} - \overset{\text{III.}}{\mathrm{W_2 C_v T_2}} + \overset{\text{II.}}{\frac{4}{3} \cdot \frac{1}{2\,g} \cdot \mathrm{W_1}\, v_1^2 \frac{1}{\mathrm{J}}}$$

$$- \overset{\text{VII + VIII.}}{\left(\frac{5}{4}\int \mathrm{P}\, d\,\mathrm{V} \, . \, \frac{1}{\mathrm{J}}\right)} = \overset{\text{IV.}}{\frac{4}{3} \cdot \frac{1}{2\,g} \mathrm{W_2}\, v_2^2 \frac{1}{\mathrm{J}}} +$$

$$\overset{\text{V.}}{(\mathrm{W_1 - W_2})\, \mathrm{C}_p \left(\frac{t_1 + t_2}{2}\right)} + \overset{\text{VI.}}{\frac{\mathrm{W_1 - W_2}}{2\,g} \, . \, v^2 \, . \, \frac{1}{\mathrm{J}}}.$$

There is one relationship in this equation that is still indefinite, namely the relationship existing between t, the temperature at the throat, at any instant, and T, the corresponding value of the absolute temperature of the gases in the cylinder. While the general mass of gas in the cylinder is expanding, as a result, on the one hand, of the increase of volume between the pistons, and, on the other hand, of the loss of gas passing out through the port, there may, or may not, be an expansion of the gas from the interior of the cylinder to the port. The relationship between t and T will depend upon the extent of the second kind of expansion. While it is to be expected that, when flow has been established, the expansion may be adiabatic, on the other hand, immediately upon the sudden opening of a port, the gases may pass out without expansion, under their molecular velocities. These two extreme cases have been considered, and the effects upon the relationship between t and T would be: (1) with adiabatic expansion,

$$t = \left(\frac{2}{\gamma + 1}\right) \mathrm{T},$$

where γ is the adiabatic index; and (2) with no expansion, $t = \mathrm{T}$. Calculations followed these two sets of conditions.

Under the conditions of pressure and temperature of these gases, the following constants are used, as representing general average values: $C_p = 0 \cdot 2852$; $C_v = 0 \cdot 2162$; $\gamma = 1 \cdot 32$; and $R = 92 \cdot 2$ ft.-lb. per lb. per deg. C. The pressure in lb. per square inch is denoted by P, and V is the volume enclosed between the pistons relative to the volume at E O (V_{EO}). The volume enclosed between the pistons at exhaust opening is 583 c.c., or 0·0206 cub. ft., and it is possible to express all corresponding volumes in terms of the volume of E O as unity. W_1, and x $(= W_1 - W_2)$ are the masses of gas, in lb.; and A is the mean port area in square feet during the step. With the aid of these constants, the equation may be converted to one in which the only unknown is $(W_1 - W_2) = x$, as follows: Using $\mathrm{P\,V = W\,R\,T}$, terms I and III are converted to the form constant. P V. Term II is always term IV of the previous step. In term IV, the velocity v_2 depends directly on $(W_1 - W_2)$, that is, on x, and inversely, on W_2, and $W_2 = (W_1 - x)$. W_2 depends directly on the volume V_2 enclosed between the pistons, and on the density of the gas. Term IV may thus be expressed in the form:

$$\text{constant} \, . \, \frac{\mathrm{V}_2^2\, x^2}{(\mathrm{W}_1 - x)}.$$

In term V, $(W_1 - W_2) = x$, and $t_1 = \text{constant} \, . \, \mathrm{T}$, that is,

$$t_1 = \text{constant} \, . \, \frac{\mathrm{P_1 V_1}}{\mathrm{W_1}};$$

t_2, similarly

$$= \text{constant} \, . \, \frac{\mathrm{P_2 V_2}}{\mathrm{W_2}},$$

so that term V can be reduced to the form:

$$\text{constant} \, . \left(\frac{\mathrm{P_1 V_1}}{\mathrm{W_1}} + \frac{\mathrm{P_2 V_2}}{\mathrm{W_1} - x}\right) x.$$

In term VI, $(W_1 - W_2) = x$, v depends directly on x, inversely on the mean density at the throat, and inversely on the area A through the port, and since it is the mean velocity, term VI can thus be reduced to the form:

$$\text{constant} \, . \left(\frac{\mathrm{V_1 V_2}}{\mathrm{W_1 V_2} + (\mathrm{W_1} - x)\, \mathrm{V_1}}\right)^2 \frac{x^3}{\mathrm{A}^2}.$$

For the case where $t = 0 \cdot 864$ T the equation becomes:

$$\overset{\text{I.}}{0 \cdot 00696\, \mathrm{P_1 V_1}} - \overset{\text{III.}}{0 \cdot 00696\, \mathrm{P_2 V_2}} + \overset{\text{II.}}{\text{Term IV of previous step.}}$$

$$\overset{\text{VII. + VIII.}}{- 1 \cdot 25 \int \mathrm{P} \, . \, d\,\mathrm{V}} = \overset{\text{IV.}}{72 \cdot 6\, \mathrm{V}_2^2 \frac{x^2}{(\mathrm{W}_1 - x)}}$$

$$+ \overset{\text{V.}}{0 \cdot 00397 \left(\frac{\mathrm{P_1 V_1}}{\mathrm{W_1}} + \frac{\mathrm{P_2 V_2}}{\mathrm{W_1} - x}\right) x} +$$

$$\overset{\text{VI.}}{1 \cdot 114 \left(\frac{\mathrm{V_1 V_2}}{\mathrm{W_1 V_1} + (\mathrm{W_1} - x)\, \mathrm{V_1}}\right)^2 \cdot \frac{x^3}{\mathrm{A}^2}}.$$

For the case where $t = \mathrm{T}$, the coefficient of term V becomes 0·00459, the other terms remaining unchanged.

(To be continued.)

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(Concluded from page 559.)

IN the previous article, the energy equation was deduced, and it will now be instructive to employ the method of tabulation. Table I, on page 618, gives the final data for the first case, in which $t = 0 \cdot 864$ T, and Table II, on the same page, gives the values for the various terms of the equation used in obtaining Table I. The first four columns in Table I were given, and the temperature and mass at E O in columns 5 and 6 were those already deduced. The values in each step of Table II were obtained by trial, and from these solutions the values in columns 7 and 8 of Table I were obtained; the values in column 6 were obtained by subtraction. From the values in column 6, the corresponding temperatures in column 5 were calculated, and from the data in columns 4 and 5, the values of the mean densities of the gases in the cylinder, given in column 9, were obtained. The mean velocities in the port were calculated from the values in column 8, and the densities under the port conditions, where the pressure is 0·544 times the pressure in the cylinder, and the temperature is 0·864 times T, were found using these values. The data in column 11 were obtained from column 7, having regard to the ratios of the mean effective areas of the port to the cross-sectional area of the cylinder. The momenta in column 12 are the products of the values in columns 7 and 10. The momenta in column 13 are the products of the values in columns 6 and 11.

Table III, on the opposite page, was obtained on the assumption that $t =$ T, and Table IV gives the corresponding values of the terms of the equation. In this case, the pressures and temperatures, and, therefore, the densities of the gases passing through the port, are taken to be the same as those in the cylinder. The data given in the various columns were otherwise deduced in the same way as in Table I.

The values of the temperature, T, and mass in the cylinder, W, of Tables I and III, are plotted in Fig. 11, page 620, on a base of degrees after E O and time. At any angle, the mass of gas in the cylinder is inversely proportional to the absolute temperature. The differences between the two cases prove to be surprisingly small, the mass of gas leaving the cylinder when additional energy is released as a result of the adiabatic expansion, being only slightly greater, leaving less gas in the cylinder. The fall of temperature is 300 deg. C. in one case, and 420 deg. C. in the other, while about 60 per cent. of the gases, in one case, and 65 per cent., in the other, have been lost from the cylinder in 22 deg., that is, in an interval of time less than 0·003 second.

Fig. 12, page 620, gives the velocities through the port, and the velocities of the centre of mass of the gases in the cylinder, plotted on a base of degrees after E O and time. It is noted that the velocity through the port begins at a value approaching 4,000 ft. per second, and then is quickly reduced to about 1,200 ft. per second, after which it rises somewhat to 1,500 ft. per second at 10 deg., then falls to a low value at 16 deg., and then rises again. When the port begins to open, there is complete discontinuity, but after motion of the external gases has begun, the outflow is no longer completely "unresisted," and lower values result. The fluctuations depend on the time-area of opening of the port, that is, on both the time rate of opening and the area at any instant, and also on the interaction between the gases leaving the cylinder and the external gases. Under the influence of the velocities through the port, and of the increasing area of opening of the port, the velocities of the centre of mass of the gases in the cylinder increase generally, with certain fluctuations during this period, and attain a high value by the end of the time. On Fig. 13, page 620, are plotted the masses of gas leaving the cylinder in each degree. These values again are the result of the two variables, velocity through the port and the area of the port. They increase to a maximum after 10 deg., then decrease to a minimum at 16 deg., and then again increase. The products of the masses and the velocities, that is, the momenta of the gases in the cylinder, and the momenta leaving the cylinder per degree, are plotted on the same base in Fig. 14, page 620. The broken curves in Figs. 13 and 14 show, respectively, the masses and the momenta passing the point P, Fig. 8, page 516, *ante*, per degree. The values on these curves were deduced from the diagrams of Fig. 9, page 516, *ante*, on the assumption of a mean value of 1,100 deg. C. abs. for the temperature. A lower temperature means, of course, an increase of the momenta passing P. The upstream Pitot tube at P was placed at the centre of the pipe, and there is evidence, not given here, that the distribution of mass over the cross-section of the pipe is not necessarily uniform. Further, the uncertainty of the values of temperature at P leads also to these values of mass and momentum not being very definite. Nevertheless, these curves give useful corroboration to the curves deduced from the cylinder diagrams, Figs. 6*a* and 6*b*, page 516, *ante*, both as regards the masses and as regards the momenta.

The state of affairs at the instant A O is such that the greater part of the cylinder gases has left the cylinder, and has passed the point P at high velocity, and thus possesses high momentum. The gases remaining in the cylinder have also acquired a considerable momentum towards the exhaust port, although their static pressure has not yet fallen to atmospheric. At A O, the admission port begins to open, and almost immediately the new charge begins to enter the cylinder, as shown on Figs. 4 and 5, page 516, *ante*. The opening of the admission port in this way naturally permits the atmospheric air to enter the cylinder behind the retreating exhaust gases, and, in this particular test, the measurements at the centre of the cylinder show at no instant a static depression in the cylinder exceeding $1\frac{1}{4}$ lb. per square inch. In this case, however, the admission gases enter the cylinder at once, and continue to enter even though there is a depression in the admission pipe, and at the admission port, for a good part of the admission period.

On the evidence deduced from the present test, therefore, the process can be described as follows: On opening the exhaust port, the time-area rate is sufficient to allow the exhaust gases to pass out with the high velocities measured, and to impart their momentum to the external gases, which, at E O, were practically at rest outside the exhaust port. As a result of this momentum, the exhaust gases and the external gases move away from the cylinder. The momentum of the gases still in the cylinder at A O, causes them to continue to move out of the cylinder, and away from the exhaust port. The effect of these momenta might be compared with that of a scavenging piston moving through the cylinder and along the exhaust pipe away from the exhaust port. There appears from the diagrams on Fig. 6*a* to be no discontinuity between the retreating exhaust gases and the new charge entering from the atmosphere through the admission port, and, during the greater part of the charging period, the pressure in the cylinder is shown as a depression of 0·5 lb. to 1·0 lb. per square inch. In a test under similar conditions, one of the two admission ports was closed, and a pressure-measuring unit was applied at the closed port. This showed that, at this point, where motion of the incoming gases was prevented, there was a static depression of 2 lb. per square inch. The depression causing the entry of the new charge through the open ports, in the present case, must therefore be regarded as a "dynamic" depression, as distinct from the static depression which can be shown to exist when the admission ports are restricted. (See also the author's letter in ENGINEERING, vol. 148, page 207, 1939).

The pressure in the cylinder rises above atmospheric at 97 deg., and this may be due to one of two causes: firstly, the atmospheric air which has entered the cylinder, and of which some has passed out through the exhaust port, may meet a resistance

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TABLE I.

1. Angle after EO. Deg.	2. Proportionate Volume. EO = 1·0.	3. Mean Effective Area during Degree. Sq. ft. × 10⁻³.	4. Indicated Pressure in Cylinder. lb. per sq. in. abs.	5. Temperature in Cylinder, T. Deg. C. abs.	6. Mass in Cylinder, W. Lb. × 10⁻².	7. Mass leaving Cylinder, $W_1 - W_2$. per deg. lb. × 10⁻⁴.	8. Mass leaving Cylinder. Lb. per sec.	9. Density under Cylinder Conditions. Lb. per cub. ft.	10. Mean Velocity through Port. Ft. per sec.	11. Mean Vertical Velocity of Mass Centre in Cylinder. Ft. per sec.	12. Momentum leaving Cylinder per deg. Lb.-ft. per sec.	13. Momentum in Cylinder. Lb.-ft. per sec.
0	1·0		74·7	1,300	0·1849			0·0898				
		0·285				0·132	0·102		4,015	16·0	0·063	0·0296
1	1·008		73·0	1,290	0·18358			0·0884				
		0·957				0·222	0·171		2,042	27·36	0·0454	0·0499
2	1·015		71·2	1,283	0·18136			0·0866				
		1·665				0·269	0·207		1,452	33·8	0·0391	0·0609
3	1·023		69·2	1,277	0·17867			0·0846				
		2·285				0·308	0·237		1,240	39·6	0·0382	0·0701
4	1·031		67·1	1,268	0·17559			0·0826				
		2·931				0·397	0·305		1,278	52·2	0·0506	0·0906
5	1·038		64·7	1,259	0·17162			0·0808				
		3·583				0·477	0·367		1,313	64·9	0·0626	0·1099
6	1·046		61·9	1,249	0·16685			0·0774				
		4·09				0·503	0·387		1,244	71·1	0·0626	0·1170
7	1·054		59·1	1,238	0·16182			0·0746				
		4·74				0·579	0·445		1,288	85·3	0·0745	0·1356
8	1·062		56·0	1,226	0·15603			0·0714				
		5·38				0·658	0·506		1,352	101·8	0·0890	0·1553
9	1·069		52·7	1,212	0·14945			0·0679				
		6·03				0·767	0·590		1,483	125·1	0·1138	0·1821
10	1·077		48·9	1,194	0·14178			0·0639				
		6·72				0·848	0·652		1,570	147·7	0·1331	0·2030
11	1·084		44·9	1,178	0·13330			0·0597				
		7·28				0·729	0·560		1,329	136·1	0·0968	0·1751
12	1·091		41·6	1,159	0·12601			0·0561				
		7·84				0·705	0·543		1,273	139·7	0·0898	0·1710
13	1·098		38·5	1,141	0·11896			0·0527				
		8·49				0·690	0·531		1,227	145·7	0·0846	0·1682
14	1·104		35·6	1,128	0·11206			0·0493				
		9·05				0·582	0·448		1,033	130·9	0·0602	0·1430
15	1·110		33·2	1,116	0·10624			0·0465				
		9·51				0·494	0·380		882	117·3	0·0435	0·1216
16	1·116		31·2	1,104	0·10130			0·0441				
		9·98				0·426	0·328		764	106·7	0·0327	0·1057
17	1·122		29·5	1,097	0·09704			0·0420				
		10·50				0·578	0·444		1,040	152·8	0·0601	0·1438
18	1·129		27·2	1,088	0·09126			0·0392				
		10·96				0·644	0·495		1,196	200·0	0·0770	0·1762
19	1·136		24·7	1,063	0·08482			0·0363				
		11·40				0·763	0·587		1,490	237·6	0·1137	0·1920
20	1·142		21·8	1,038	0·07719			0·0328				
		11·79				0·568	0·436		1,174	193·6	0·0667	0·1430
21	1·148		19·8	1,021	0·07151			0·0302				
		12·09				0·643	0·494		1,418	240·0	0·0911	0·1638
22	1·154		17·5	998	0·06508			0·0274				

TABLE II.

Angle after EO. Deg.	I.	+ II.	− III.	− (VII + VIII).	− Left Hand Side.	IV.	+ V.	+ VI.	− Right Hand Side.
	(Expressed in Centigrade Heat Units.)								
0									
	0·5197	0	0·5119	0·001215	0·00659	Negl.	0·004212	0·002389	0·00655
1									
	0·5119	Negl.	0·5028	0·00103	0·00807	0·000020	0·007018	0·001022	0·00806
2									
	0·5028	0·000020	0·4926	0·0011	0·00912	0·000031	0·008482	0·000634	0·00914
3									
	0·4926	0·000031	0·4813	0·0011	0·01023	0·000042	0·009656	0·000524	0·01022
4									
	0·4813	0·000042	0·4673	0·0009	0·01315	0·000072	0·012370	0·000718	0·01316
5									
	0·4673	0·000072	0·4506	0·0010	0·01577	0·000109	0·014750	0·000921	0·01578
6									
	0·4506	0·000109	0·4334	0·0009	0·01641	0·000126	0·015425	0·000869	0·01642
7									
	0·4334	0·000126	0·4138	0·0009	0·01883	0·000178	0·017596	0·001068	0·01884
8									
	0·4138	0·000178	0·3919	0·0007	0·02138	0·000240	0·019800	0·001337	0·02138
9									
	0·3919	0·000240	0·3664	0·0007	0·02504	0·000351	0·022780	0·001682	0·02501
10									
	0·3664	0·000351	0·3386	0·0006	0·02755	0·000461	0·024775	0·002302	0·02754
11									
	0·3386	0·000461	0·3158	0·0005	0·02276	0·000365	0·020965	0·001434	0·02276
12									
	0·3158	0·000365	0·2942	0·0003	0·02167	0·000366	0·020010	0·001270	0·02165
13									
	0·2942	0·000366	0·2734	0·0003	0·02087	0·000377	0·019315	0·001165	0·02086
14									
	0·2734	0·000377	0·2564	0·0003	0·01708	0·000285	0·016090	0·000690	0·01707
15									
	0·2564	0·000285	0·2423	0·0002	0·01419	0·000217	0·013520	0·000426	0·01416
16									
	0·2423	0·000217	0·2303	0·0002	0·01202	0·000172	0·011570	0·000282	0·01202
17									
	0·2303	0·000172	0·2137	0·0002	0·01657	0·000341	0·015540	0·000709	0·01659
18									
	0·2137	0·000341	0·1953	0·0002	0·01854	0·000458	0·017050	0·001022	0·01853
19									
	0·1953	0·000458	0·1733	0·0001	0·02236	0·000715	0·019765	0·001882	0·02236
20									
	0·1733	0·000715	0·1582	0·0001	0·01572	0·000432	0·014420	0·000871	0·01572
21									
	0·1582	0·000432	0·1405	0·0001	0·01807	0·000616	0·016015	0·001446	0·01808
22									

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TABLE III.

1. Angle after EO. Deg.	2. Proportionate Volume. EO = 1·0.	3. Mean Effective Area during Degree. Sq. ft. × 10^{-3}.	4. Indicated Pressure in Cylinder. Lb. per sq. in. abs.	5. Temperature in Cylinder. T. Deg. C. abs.	6. Mass in Cylinder, W. Lb. × 10^{-2}	7. Mass leaving Cylinder, $W_1 - W_2$. Per deg. lb. × 10^{-4}.	8. Mass leaving Cylinder. Lb. per sec.	9. Density under Cylinder Conditions. Lb. per cub. ft.	10. Mean Velocity through Port. Ft. per sec.	11. Mean Vertical Velocity of Mass Centre in Cylinder. Ft. per sec.	12. Momentum leaving Cylinder per deg. Lb.-ft. per sec.	13. Momentum in Cylinder. Lb.-ft. per sec.
0	1·0	0·235	74·7	1,300	0·1849	0·124	0·0954	0·0898	3,758	14·97	0·0466	0·0276
1	1·008	0·957	73·0	1,289	0·18366	0·200	0·1539	0·0885	1,833	24·75	0·0383	0·0451
2	1·015	1·665	71·2	1,280	0·18166	0·238	0·1831	0·0869	1,279	29·8	0·0304	0·0538
3	1·023	2·285	69·2	1,271	0·17928	0·272	0·209	0·0850	1,089	34·8	0·0296	0·0619
4	1·031	2·931	67·1	1,261	0·17656	0·352	0·271	0·0831	1,128	46·2	0·0397	0·0808
5	1·038	3·533	64·7	1,250	0·17304	0·425	0·327	0·0808	1,167	57·6	0·0495	0·0985
6	1·046	4·09	61·9	1,236	0·16879	0·447	0·344	0·0781	1,093	62·5	0·0490	0·1040
7	1·054	4·74	59·1	1,221	0·16432	0·519	0·390	0·0756	1,138	75·4	0·0500	0·1218
8	1·062	5·38	56·0	1,203	0·15913	0·594	0·457	0·0727	1,194	89·9	0·0709	0·1403
9	1·069	6·03	52·7	1,184	0·15319	0·700	0·539	0·0697	1,319	111·1	0·0923	0·1662
10	1·077	6·72	48·9	1,159	0·14619	0·778	0·599	0·0659	1,393	131·0	0·1083	0·1867
11	1·084	7·28	44·9	1,132	0·13841	0·668	0·514	0·0619	1,171	119·1	0·0782	0·1610
12	1·091	7·84	41·6	1,110	0·13173	0·650	0·500	0·0586	1,119	122·8	0·0727	0·1570
13	1·098	8·49	38·5	1,087	0·12523	0·641	0·494	0·0554	1,082	128·6	0·0695	0·1568
14	1·104	9·05	35·6	1,065	0·11882	0·542	0·417	0·0522	906	114·7	0·0491	0·1332
15	1·110	9·51	33·2	1,046	0·11340	0·461	0·355	0·0496	771	102·7	0·0358	0·1140
16	1·116	9·98	31·2	1,031	0·10879	0·398	0·306	0·0472	662	92·4	0·0264	0·0986
17	1·122	10·60	29·5	1,016	0·10481	0·548	0·421	0·0454	912	134·0	0·0500	0·1369
18	1·129	10·96	27·2	995	0·09933	0·618	0·475	0·0427	1,050	175·7	0·0649	0·1690
19	1·136	11·40	24·7	970	0·09315	0·743	0·571	0·0398	1,315	209·8	0·0957	0·1873
20	1·142	11·79	21·8	936	0·08572	0·554	0·426	0·0364	1,028	169·3	0·0569	0·1403
21	1·148	12·09	19·8	911	0·08018	0·638	0·401	0·0340	1,247	210·6	0·0795	0·1622
22	1·154		17·5	880	0·07380			0·0311				

TABLE IV.

Angle after EO. Deg.	I.	+ II.	− III.	− (VII + VIII).	= Left Hand Side.	IV.	+ V.	+ VI.	= Right Hand Side.
	(Expressed in Centigrade Heat Units.)								
0	0·5197	0	0·5119	0·001215	0·00658	Negl.	0·004579	0·001962	0·00654
1	0·5119	Negl.	0·5028	0·00103	0·00807	0·000017	0·007328	0·000747	0·00809
2	0·5028	0·000017	0·4926	0·0011	0·00912	0·000024	0·008060	0·000430	0·00911
3	0·4926	0·000024	0·4813	0·0011	0·01022	0·000032	0·009825	0·000358	0·01022
4	0·4813	0·000032	0·4673	0·0009	0·01313	0·000056	0·01261	0·000496	0·01316
5	0·4673	0·000056	0·4506	0·0010	0·01576	0·000085	0·01506	0·000640	0·01578
6	0·4506	0·000085	0·4334	0·0009	0·01638	0·000098	0·01566	0·000594	0·01635
7	0·4334	0·000098	0·4138	0·0009	0·01880	0·000130	0·01795	0·000742	0·01883
8	0·4138	0·000130	0·3919	0·0007	0·02134	0·000191	0·02022	0·000952	0·02136
9	0·3919	0·000191	0·3664	0·0007	0·02499	0·000282	0·02339	0·001350	0·02502
10	0·3664	0·000282	0·3386	0·0006	0·02748	0·000373	0·02543	0·001678	0·02748
11	0·3386	0·000373	0·3158	0·0005	0·02267	0·000292	0·02136	0·001020	0·02267
12	0·3158	0·000292	0·2942	0·0003	0·02159	0·000296	0·02037	0·000900	0·02157
13	0·2942	0·000296	0·2734	0·0003	0·02080	0·000306	0·01968	0·000835	0·02082
14	0·2734	0·000306	0·2564	0·0003	0·01701	0·000232	0·01631	0·000494	0·01704
15	0·2564	0·000232	0·2423	0·0002	0·01413	0·000177	0·01366	0·000304	0·01414
16	0·2423	0·000177	0·2303	0·0002	0·01198	0·000138	0·01162	0·000194	0·01195
17	0·2303	0·000138	0·2137	0·0002	0·01654	0·000280	0·01572	0·000509	0·01651
18	0·2137	0·000280	0·1953	0·0002	0·01848	0·000384	0·01732	0·000762	0·01847
19	0·1953	0·000384	0·1733	0·0001	0·02228	0·000611	0·02025	0·001430	0·02220
20	0·1733	0·000611	0·1582	0·0001	0·01561	0·000367	0·01461	0·000649	0·01563
21	0·1582	0·000367	0·1405	0·0001	0·01797	0·000535	0·01632	0·001112	0·01797
22									

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Fig. 11.

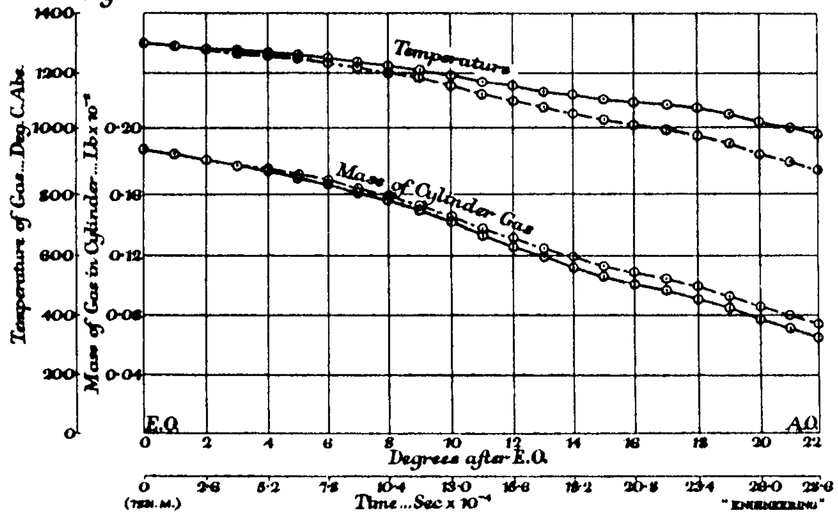


Fig. 12.

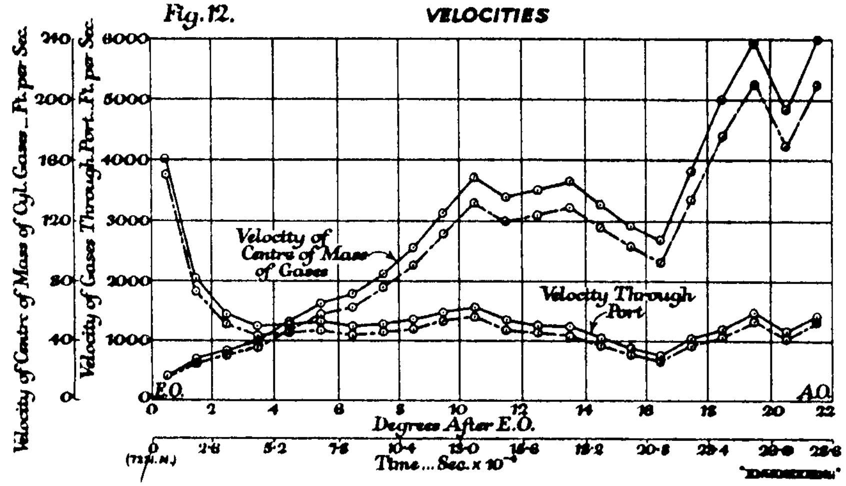


Fig. 13.

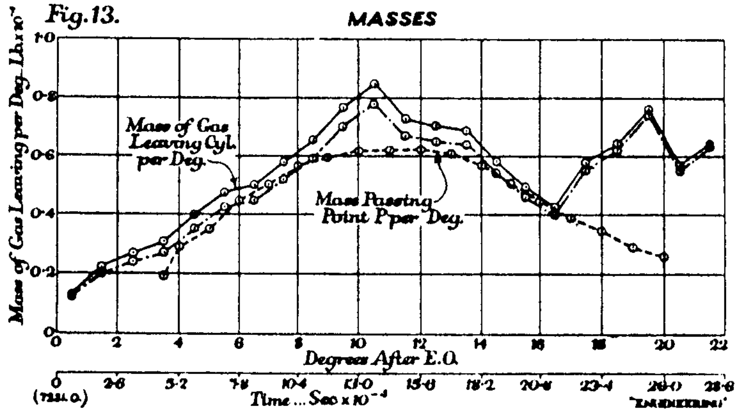
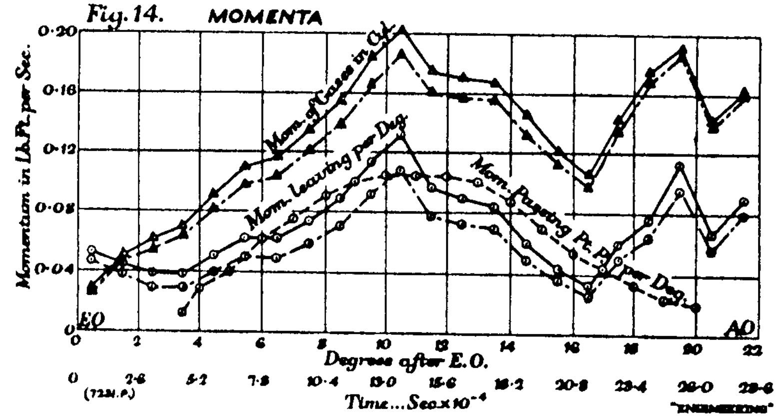


Fig. 14.



in the pipe outside, such as is indicated in Fig. 9, page 516, *ante*. On being decelerated, the new charge tends to pile itself up in the cylinder and in the exhaust pipe, with consequent increases of pressure. The exhaust port is, however, gradually closing, and there is no effect from outside this port after 104 deg. Secondly, the momentum of the exhaust gases, and of the external gases to which it has been communicated, will be dissipated more and more as greater and greater masses of external gas come under its influence, and there is evidence of the progress of a return motion of the gases towards the exhaust port. In some combinations of circumstances, the latter may predominate over the former, and may supercharge the cylinder backwards through the exhaust port, the new air which had previously passed out through the exhaust port being forced backwards into the cylinder. Concurrently with both these actions there is the fact that the pistons are now moving together again, and the volume enclosed between them is being reduced.

Before drawing conclusions from this analysis it is perhaps desirable to point out certain respects in which, in the author's view, the measurements given above could profitably be supplemented. First of all, it was found practicable to take pressure

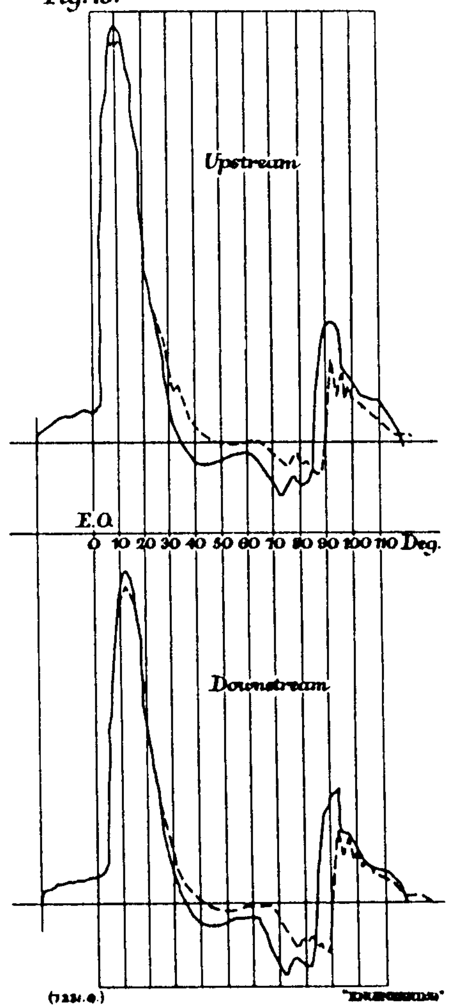
measurements at only one point in the cylinder, namely, at the wall near the middle. The distribution of static pressure, velocity, temperature and density, that is, the distribution of the mass of the gases in the cylinder, remains unmeasured, and the simplest treatment appeared to be to take the velocity distribution as linear from top to bottom, and to assume the temperature, pressure and density as uniform in general. In the case with adiabatic expansion, given in Tables I and II, there are, of course, superimposed on this generally uniform distribution, corresponding changes of pressure, temperature and density in an indefinite, but restricted, volume from the cylinder to the port. As stated earlier, the lack of means for measuring even the mean densities or temperatures in the gases rendered necessary an indirect deduction of the temperature.

In the treatments based on continuous flow, the "throat" of the convergent-divergent nozzle is taken as the port opening, and a suitable coefficient of discharge is assumed. The author has followed this method in both the cases of Tables I and III, and has taken 0·8 throughout as the value of the coefficient. In view of the pressures recorded at P in Fig. 9, page 516, *ante*, however, the evidence in support of the use of this assumption in the present case is very uncertain. Measurements at the exhaust port, unfortunately impracticable, might have thrown light on this matter. In these treatments, too, the whole of the energy to produce the velocity through the port is calculated from that available from the adiabatic expansion of the gases from the cylinder to the port. Between the values of velocity in Tables I and III there is little difference, the increase of velocity following the adiabatic expansion being small. All this is consistent with the view expressed earlier, that treatments based on continuous flow are quite incomplete in connection with the phenomena in actual engines.

Altogether, the author feels that, with regard to the properties of gases, the knowledge applied at present to internal-combustion engines is very incomplete. This is a fundamental matter in which the engineer may justifiably expect more guidance from the physicist, in order to fill up such gaps as those indicated above. In considering the present experimental treatment, the author finds further satisfactory support for the opening statement of his earlier report, namely:

"The basic contribution that Kadenacy has made to engine design is his discovery that, immediately on rapidly opening the exhaust ports of an engine during the expansion stroke, there is, within the first interval of time for a few thousandths of a second, an urge or impulse in the gases within the cylinder to escape very rapidly from the cylinder, leaving behind them a depression. By suitable timing of the admission valve or port, the new charge is arranged to enter the cylinder behind the retreating exhaust gas. These actions are found to occur with suitable port design and timing, and no

Fig. 15.



scavenging pumps, inlet pipes or exhaust pipes are necessary to enable such an engine to run and deliver power. This extremely rapid exit of the exhaust gases may conveniently be termed ballistic, and experiment shows that it depends only on time."

The present analysis demonstrates that this so-called "ballistic" exit of the gases is the dominating factor in establishing the conditions leading to the direct charging of the cylinder, in this case from the atmosphere, through the admission port. The kind of interaction occurring between the gases leaving the exhaust port and the external gases is shown in Fig. 9, page 516, *ante*, and the exhaust organ must facilitate the exit of the gases from the cylinder. This organ must be dimensioned to meet the practical conditions of running, that is, so that the charging will give the necessary order of B.M.E.P. over the speed range desired.

Appendix.

In describing the exhaust system shown in Fig. 8, page 516, *ante*, reference was made to the supplementary volume shown after the measuring point P, and it was stated that, for the purposes of the present analysis, the effect of this extra volume may be neglected. Fig. 15, on this page, shows two pairs of diagrams taken at a point 9¼ in. from the exhaust port, and thus beyond the position of the supplementary volume. One pair are upstream diagrams and the other downstream diagrams, taken from this point, with the engine running at 1,200 r.p.m., and developing a B.M.E.P. of 99·7 lb. per square inch. The diagrams show pressure on a base of crank angle after E O, but the pressures were not calibrated and thus no scale is given. The broken lines show the conditions with, and the firm lines without, the supplementary volume. It is noticed that the curves follow practically the same course in both pairs for the first 24 deg. after E O. Ignoring the interval of time between action at the exhaust port and action at this measuring point, the differences shown, and occurring subsequently, can have no effect on the motion of the gases passing through the exhaust port during the first 22 deg. from E O, that is, during the interval dealt with in Tables I to IV. While the later differences are important in relation to the general engine performance, they, therefore, do not enter into the present analysis.

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AUG. 16, 1940. p.134

RAPID DISCHARGE OF GAS FROM A VESSEL INTO THE ATMOSPHERE.

By E. GIFFEN, Ph.D., M.I.Mech.E.

Part I. *Based on Consideration of Wave Action in the Vessel.*—The problem considered in the following analysis is the rate at which the internal pressure decreases when the contents of a vessel are allowed to discharge into the atmosphere through a rapidly-opened port. This has an important bearing upon the exhaust process in internal-combustion engines, particularly in the case of two-stroke cycle engines operating at high speeds. In such engines, the time available for the discharge of the exhaust gas is very short and requires the rapid opening of the exhaust port or valve. The rate of opening is controlled either by the movement of the piston or by a cam driven from the engine crankshaft, but in either case the rate of variation of the port area is accurately known. Further, the movement of the piston is comparatively small during the period of pressure release, so that the cylinder may be regarded as being of constant volume during this time; it will be seen later, however, from the method of treatment adopted, that any known rate of variation of the volume of the vessel would present no difficulty. While the conditions of the problem discussed resemble, to some extent, those of the exhaust period in an internal-combustion engine, the two problems are not exactly similar, since in the case of the engine a pipe is usually interposed between the cylinder and the outlet to the atmosphere. The presence of the pipe has a quite definite effect upon the conditions inside the cylinder, due, firstly, to the resistance offered by the contents of the pipe to the discharging gas, and secondly, to the wave action set up in the pipe. For the present, however, this additional complication is avoided by directing attention to the basic problem of discharge directly into the atmosphere.

It is necessary at this point to consider the implications of the term "rapid" as applied either to the opening of the discharge port or to the discharge of the gas itself. The term is only a relative one, and involves the comparison of the rates at which the various changes occur with some condition which plays a controlling part in the discharge. This controlling condition is the speed with which any disturbance is transmitted through the contents of the vessel, *i.e.*, the speed of sound in the gas, and it has an important influence upon the phenomena occurring inside the vessel during the discharge process. When the orifice or port is small in relation to the size of the vessel, the rate of variation of the pressure inside the vessel is small compared with the rate at which the variations in pressure are transmitted through the gas, and the pressure may be assumed to be constant in all parts of the vessel. Under these conditions, also, the velocity with which the gas in the vessel approaches the discharge port is relatively small, and is usually neglected. If the size of the discharge port be increased, the rate of discharge, and the rate of variation of pressure inside the vessel, will also increase. A point will be reached when the rate of change of pressure near the discharge port is so great that an appreciable pressure difference will exist in different parts of the vessel, on account of the time required for the transmission of any pressure change from one part to another. There will, therefore, be a pressure gradient in the vessel, those points farthest removed from the port lagging most behind the pressure at the port. The rate of discharge, however, is determined, so far as the conditions inside the vessel are concerned, by the pressure and density close to the port, and by the velocity with which the gas approaches the port. This velocity of approach follows, of course, from the pressure variations throughout the vessel. It will be seen, therefore, that the conditions near the discharge port may vary considerably from those which follow from the assumption of a uniform pressure and zero velocity inside the vessel, and the calculated rate of discharge would be affected accordingly. The factors that determine the extent of the pressure variations in the vessel are: firstly, the rate of change of the pressure near the port; and secondly, the time required for any change at the port to be transmitted throughout the vessel. The former depends upon the size of the discharge port in relation to that of the vessel, as well as upon the pressure difference on the two sides of the port; and the latter depends upon the distance from the port to the farthest point in the vessel, and also upon the speed of sound in the gas. When either of these factors produces such variations in the internal conditions that they have an appreciable effect upon the calculated rate of discharge, or lead to phenomena which would be absent with a uniform pressure in the vessel, the discharge may be said to be "rapid." In such cases it becomes necessary to consider the mechanism whereby the variations in pressure near the port are transmitted through the contents of the vessel.

Wave Action in the Vessel.—In a consideration of the propagation of these pressure waves, it is convenient to assume the vessel cylindrical in shape, with the discharge port at one end, so that any disturbance at the port is transmitted as a plane wave along the axis of the cylinder. The properties of such waves have been discussed by Earnshaw (1860)* and by Rayleigh (1910)†, and for the present purpose the most important results may be stated briefly as follows: If a plane wave of finite amplitude passes through a column of gas originally at rest and having a density ρ_0, then at any point where the density is ρ_1 the gas will have a velocity v_1, in the direction of the discharge port, given, for adiabatic conditions, by the relation:—

$$\left(\frac{\rho_1}{\rho_0}\right)^{\frac{\gamma-1}{2}} = 1 \pm \frac{\gamma-1}{2}\frac{v_1}{c_0}, \quad . \quad . \quad (1)$$

where γ is the adiabatic index, and c_0 the speed of sound in the gas in the initial condition. The positive and negative signs apply to positive and negative waves, respectively, *i.e.*, to waves in which the direction of propagation of the wave is in the same direction as, or in the opposite direction to, the movement of the gas. The speed of propagation varies with the density, *i.e.*, with the gas velocity, and is given by the expression $c_0 \pm \frac{\gamma+1}{2} \cdot v_1$.

The positive and negative signs again refer to positive and negative waves.

If pressure be substituted for density in (1), the equation becomes, for adiabatic conditions,

$$\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{2\gamma}} = 1 \pm \frac{\gamma-1}{2} \cdot \frac{v_1}{c_0}, \quad . \quad . \quad (2)$$

which is a more convenient form for the present work. In this expression, p_1 is the absolute pressure in the vessel at the point where the gas has the velocity v_1, and p_0 the absolute pressure at the point where the gas is at rest, before disturbance by the pressure wave.

With these results it is now possible to follow the various changes in pressure and velocity inside a cylindrical vessel when discharge takes place through a port in one end. The gas is initially at rest and at a pressure denoted by p_0. It is assumed that the opening of the port occupies a very short but finite interval of time, and that the area of the port then remains constant. As soon as this constant area has been reached the pressure inside the vessel, near the port, will have fallen to p_1, and this drop in pressure is propagated as a negative wave along the vessel, setting the gas in motion towards the port. Fig. 1 shows the pressure inside the vessel at an instant when the wave has traversed about half the length of the vessel. In the portion on the left, in which the pressure is p_0, the gas is still at rest. On the right, where the pressure has fallen to p_1 by the passage of the wave, the gas is flowing towards the orifice with a velocity v_1, given by the relation expressed in equation (2), using the negative sign. The time represented by the sloping line from p_0 to p_1 is the time required for the port to open. The value of p_1 will be

* Earnshaw, *Phil. Trans. Roy. Soc.*, vol. 150, page 133 (1860).

† Rayleigh, *Proc. Roy. Soc.* (A), vol. 84, page 247 (1910).

gas behind it is $\frac{p_1}{\rho_1}$; similarly, the work done by unit mass at the plane B B is $\frac{p_2}{\rho_2}$. The net external work done by unit mass of gas is therefore $\frac{p_2}{\rho_2} - \frac{p_1}{\rho_1}$. The internal energies of unit mass of gas at planes A A and B B are J C_v T_1 and J C_v T_2, respectively, J being the mechanical equivalent of heat, C_v the specific heat at constant volume, and T the absolute temperature. The kinetic energies are $\frac{v_1^2}{2g}$ and $\frac{v_2^2}{2g}$ at the two planes. These quantities, when substituted in the equation representing the conservation of energy, give, for adiabatic conditions,

$$\frac{p_2}{\rho_2} - \frac{p_1}{\rho_1} + \mathrm{J}\, C_v (T_2 - T_1) + \frac{1}{2\,g}(v_2^2 - v_1^2) = 0,$$

which reduces to the usual form

$$v_2^2 = v_1^2 + 2\,g\,\frac{\gamma}{\gamma - 1}\cdot\frac{p_1}{\rho_1}\left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}. \quad (3)$$

It is well known that, for flow under these conditions, the mass flow through a discharge orifice reaches a maximum when the velocity at the minimum section of the jet reaches the speed of sound at the prevailing pressure and density. Further, if the initial velocity v_1 be zero, the pressure p_2 at this minimum section is related to the initial pressure p_1 by the equation

$$\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma + 1}.$$

In the present problem the velocity v_1 is not negligible, and when this is taken into account it may readily be shown that the critical pressure ratio takes the form

$$\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma + 1} + \frac{\gamma - 1}{\gamma + 1}\cdot\frac{v_1^2}{c_1^2}. \quad (4)$$

Here c_1 is the speed of sound in the gas at the initial condition, *i.e.*,

$$c_1 = \sqrt{\frac{g\,\gamma\,p_1}{\rho_1}}.$$

The exit velocity is then equal to the speed of sound at the exit conditions, or

$$v_2 = c_2 = \sqrt{\frac{g\,\gamma\,p_2}{\rho_2}}.$$

When the atmospheric pressure p_a is less than the value of p_2 as given by equation (4), the rate of discharge is independent of p_a; the value of p_2 in equation (3) is then that derived from the critical pressure ratio, and the value of v_2 is the speed of sound. As p_1 decreases, the exit pressure p_2 also decreases, until it reaches that of the atmosphere. For lower values of p_1 the atmospheric pressure p_a is substituted for p_2 in equation (3), and the exit velocity v_2 is then less than the speed of sound. It may be noted here that equation (3) is based on the assumption that the mass flow is the same at the planes A A and B B in Fig. 3. This is true when the pressures at these planes remain constant in time, as is the case during each of the intervals into which the total discharge period is divided. When the pressures vary, a small time interval is required for a disturbance at one plane to travel to the other, and if the planes are far apart, this time lag may have to be taken into account in deriving the equation for the final velocity. In the present problem the planes A A and B B may be chosen immediately before and after the discharge port, so close together, when compared with the length of the vessel, that the time for a change at one plane to reach the other is negligible. The mass flow may then be taken as the same at the two planes, and equation (3) is directly applicable to the discharge conditions, even when the pressure varies rapidly.

The initiation of the flow, at the instant when the port begins to open, requires some consideration, especially from the point of view of the resistance to flow offered by the gas outside the port. If the discharge were taking place into a pipe, the issuing gas would set up a pressure wave in the gas in the pipe. The gas flowing from the vessel might be considered to act as a piston, communicating its velocity to the gas in the end of the pipe; the accompanying variations of pressure in the pipe follow from equation (2), and the pressure wave is propagated along the pipe in the manner already described. The rate at which the gas begins to issue from the discharge port, and the manner in which the discharge increases, therefore, control the form of the wave produced in the pipe. In practice, however, the rate of discharge increases continuously from zero, since flow begins as soon as any finite area of discharge port is available, and in the problem considered the port area increases progressively from zero. Since any disturbance is transmitted through the gas with the speed of sound, the effect of any variation in the port area is communicated to the neighbouring gas with this speed; hence, unless the port opens with a speed greater than that of sound, variations in the area of the port will be instantaneously accompanied by variations in the rate of discharge. During the initiation of the flow, therefore, the pressure in the wave produced in the pipe increases continuously from zero. When the gas discharges into the atmosphere instead of into a pipe, a similar process takes place, but the pressure generated by the escaping gas is propagated outwards in all directions as a spherical wave. For a given rate of discharge, the increase in pressure in the spherical wave is very much less than in the plane wave produced in the pipe, and is, in fact, so small that its effect in resisting the discharge may be neglected. The important point, however, is that, for discharge into a pipe or into the atmosphere, so long as the rate of opening of the port is less than the speed of sound, there is no question of any discontinuity at the initiation of the discharge, and equation (3) applies to the discharge conditions from the instant at which the port begins to open. In the present discussion, no account is taken of the effect of the pressure wave generated in the atmosphere in resisting the discharge from the vessel; it is assumed that the pressure outside the vessel remains constant and equal to that of the atmosphere. As pointed out above, this resistance offered by the gas outside the port is more important in the case when a pipe is fitted to the discharge port.

(To be continued.)

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RAPID DISCHARGE OF GAS FROM A VESSEL INTO THE ATMOSPHERE.

By E. GIFFEN, Ph.D., M.I.Mech.E.

(Continued from page 136.)

Determination of p_1 during any Interval.—Two cases must be considered, the first being when the internal pressure is above that corresponding to the critical condition at the port, and the second when the internal pressure is below the critical pressure. Taking the first case, the pressure ratio $\frac{p_a}{p_1}$ is less than the critical pressure ratio, so that the exit pressure p_2 is unknown. At the beginning of the interval, the pressure inside the vessel is uniform and equal to p_0, and the gas is at rest. A negative wave is produced, giving the gas a forward velocity v_1, while the pressure falls to p_1; both v_1 and p_1 are unknown. The three equations from which p_1, v_1 and p_2 may be determined are :—(*a*) equation (2), page 135, *ante*, relating to the wave inside the vessel;

(*b*) equation (4), page 136, *ante*, giving the value of the critical pressure ratio; (*c*) the equation representing continuity of mass flow at the two planes A A and B B in Fig. 3, page 135, *ante*. This equation is

$$\rho_1 v_1 a_1 = \rho_2 v_2 a_2 \quad . \quad . \quad . \quad . \quad (5)$$

Also, since all operations are assumed to be adiabatic, the general relation, $\frac{p}{\rho^{\gamma}} =$ constant, applies throughout, *i.e.*,

$$\frac{p_0}{\rho_0^{\gamma}} = \frac{p_1}{\rho_1^{\gamma}} = \frac{p_2}{\rho_2^{\gamma}}.$$

In a given problem the pressure p_0 at the beginning of any interval is known, from the calculations for the preceding interval; the areas a_1 and a_2 are also known. The densities, ρ_0 and ρ_1, and the speeds of sound, c_0 and c_1, may be expressed in terms of the pressures p_0 and p_1; the final velocity is equal to the speed of sound at the exit condition, and may be expressed in terms of p_2. These substitutions take the form:—

$$c_0 = \sqrt{\frac{\gamma g p_0}{\rho_0}};$$

$$c_1 = \sqrt{\frac{\gamma g p_0^{\frac{1}{\gamma}}}{\rho_0}} \cdot p_1^{\frac{\gamma-1}{2\gamma}};$$

$$v_2 = c_2 = \sqrt{\frac{\gamma g p_0^{\frac{1}{\gamma}}}{\rho_0}} \cdot p_2^{\frac{\gamma-1}{2\gamma}};$$

$$\rho_1 = \frac{\rho_0}{p_0^{\frac{1}{\gamma}}} \cdot p_1^{\frac{1}{\gamma}};$$

$$\rho_2 = \frac{\rho_0}{p_0^{\frac{1}{\gamma}}} \cdot p_2^{\frac{1}{\gamma}}.$$

Equation (2) gives

$$v_1 = \left(p_0^{\frac{\gamma-1}{2\gamma}} - p_1^{\frac{\gamma-1}{2\gamma}}\right)\frac{2}{\gamma-1}\sqrt{\frac{\gamma g p_0^{\frac{1}{\gamma}}}{\rho_0}}.$$

When this expression for v_1 and the other substitutions are inserted in equation (4) this gives

$$p_2^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma+1} \cdot p_1^{\frac{\gamma-1}{\gamma}}$$

$$+ \frac{4}{(\gamma+1)(\gamma-1)}\left(p_0^{\frac{\gamma-1}{\gamma}} + p_1^{\frac{\gamma-1}{\gamma}} - 2p_0^{\frac{\gamma-1}{2\gamma}} p_1^{\frac{\gamma-1}{2\gamma}}\right) \quad . \quad . \quad (6)$$

With the same substitutions equation (5) may be written

$$p_2^{\frac{\gamma+1}{2\gamma}} = \frac{2}{\gamma-1} \cdot \frac{a_1}{a_2} \cdot p_1^{\frac{1}{\gamma}}\left(p_0^{\frac{\gamma-1}{2\gamma}} - p_1^{\frac{\gamma-1}{2\gamma}}\right).$$

Both sides of this equation being raised to the power $\frac{2(\gamma-1)}{\gamma+1}$, then

$$p_2^{\frac{\gamma-1}{\gamma}} = \left\{\frac{2}{\gamma-1} \cdot \frac{a_1}{a_2}\left(p_0^{\frac{\gamma-1}{2\gamma}} p_1^{\frac{1}{\gamma}} - p_1^{\frac{\gamma+1}{2\gamma}}\right)\right\}^{\frac{2(\gamma-1)}{\gamma+1}} \quad . \quad . \quad (7)$$

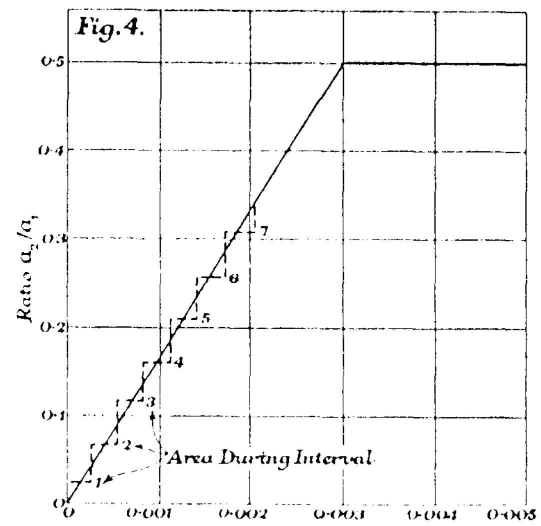
The two expressions for $p_2^{\frac{\gamma-1}{\gamma}}$, in (6) and (7), may be equated, giving

$$\left\{\frac{2}{\gamma-1} \cdot \frac{a_1}{a_2}\left(p_0^{\frac{\gamma-1}{2\gamma}} p_1^{\frac{1}{\gamma}} - p_1^{\frac{\gamma+1}{2\gamma}}\right)\right\}^{\frac{2(\gamma-1)}{\gamma+1}}$$

$$+ \frac{8}{(\gamma+1)(\gamma-1)} p_0^{\frac{\gamma-1}{2\gamma}} p_1^{\frac{\gamma-1}{2\gamma}} - \frac{2}{\gamma-1} p_1^{\frac{\gamma-1}{\gamma}}$$

$$= \frac{4}{(\gamma+1)(\gamma-1)} p_0^{\frac{\gamma-1}{\gamma}} \quad . \quad . \quad (8)$$

Here the only unknown is p_1, and this is most readily found by trial. A few values of p_1 are chosen, in the neighbourhood of the expected value, and the left-hand side of equation (8) is evaluated and plotted against p_1. The point on the curve corresponding to the right-hand side of the equation gives the value of p_1 for that particular interval. When p_1 is determined, v_1 may be calculated from equation (2), using the negative sign. After reflection at the closed end, the wave moves forward, leaving the gas again at rest. The pressure p_0^1 in the still gas is then found from equation (2), but with the positive sign. This pressure is the initial pressure p_0 for the next interval, when another negative wave passes through the gas, giving another value of p_1 at the discharge end of the vessel, and a new value of v_1 for the velocity



with which the gas approaches the port. This process of calculation is repeated until the pressure inside the vessel has fallen to a value such that $\frac{p_a}{p_1}$ is greater than the critical pressure ratio; a new set of conditions then apply to the rate of discharge.

Considering next the second case, when the internal pressure is below the critical pressure, equations (2) and (5) still apply to the problem, but instead of (4) the general equation (3) must now be used. Under the conditions considered in the previous section, the final pressure p_2 was unknown, and was eliminated from the equations. In the present case, p_2 is constant and equal to p_a, the atmospheric pressure; the final velocity v_2 is now the unknown quantity.

Making the same substitutions as before for ρ_1, ρ_2 and v_1 in equation (3), this becomes

$$c_2^2 = \frac{2g\gamma}{\gamma-1} \cdot \frac{p_0^{\frac{1}{\gamma}}}{\rho_0}\left\{\frac{2}{\gamma-1}\left(p_0^{\frac{\gamma-1}{2\gamma}} - p_1^{\frac{\gamma-1}{2\gamma}}\right)^2 + p_1^{\frac{\gamma-1}{\gamma}} - p_a^{\frac{\gamma-1}{\gamma}}\right\} \quad . \quad . \quad (9)$$

Also equation (5) takes the form

$$v_2 = \frac{2}{(\gamma-1) p_a^{\frac{1}{\gamma}}}\sqrt{\frac{g\gamma p_0^{\frac{1}{\gamma}}}{\rho_0}} \cdot \frac{a_1}{a_2} p_1^{\frac{1}{\gamma}}\left(p_0^{\frac{\gamma-1}{2\gamma}} - p_1^{\frac{\gamma-1}{2\gamma}}\right),$$

so that

$$c_2^2 = \frac{4}{(\gamma-1)^2 p_a^{\frac{2}{\gamma}}} \cdot \frac{g\gamma p_0^{\frac{1}{\gamma}}}{\rho_0} \cdot \left(\frac{a_1}{a_2}\right)^2 \cdot p_1^{\frac{2}{\gamma}}\left(p_0^{\frac{\gamma-1}{2\gamma}} - p_1^{\frac{\gamma-1}{2\gamma}}\right)^2 \quad . \quad . \quad (10)$$

These two values of c_2^2 in (9) and (10) are equated, leaving p_1 as the only unknown quantity. The equation takes the form

$$\frac{2}{(\gamma-1) p_a^{\frac{2}{\gamma}}} \cdot \left(\frac{a_1}{a_2}\right)^2\left(p_0^{\frac{\gamma-1}{\gamma}} p_1^{\frac{2}{\gamma}} - 2p_0^{\frac{\gamma-1}{2\gamma}} p_1^{\frac{\gamma+3}{2\gamma}} + p_1^{\frac{\gamma+1}{\gamma}}\right)$$

$$+ \frac{4}{\gamma-1} \cdot p_0^{\frac{\gamma-1}{2\gamma}} p_1^{\frac{\gamma-1}{2\gamma}} - \frac{\gamma+1}{\gamma-1} p_1^{\frac{\gamma-1}{\gamma}}$$

$$= \frac{2}{\gamma-1} p_0^{\frac{\gamma-1}{\gamma}} - p_a^{\frac{\gamma-1}{\gamma}} \quad . \quad . \quad . \quad (11)$$

From this equation p_1 is determined as before by trial; v_1 may then be found from equation (2) and also p_0^1, the pressure after reflection, as explained previously. This process is repeated for successive intervals until the pressure after reflection reaches that of the atmosphere, or falls below it.

Application to Typical Problem.—The results obtained in the previous section will now be applied to the problem of discharge from a vessel of constant volume, from given values of the initial pressure and density, through a port for which the rate of opening is known. It is required to trace the variation of pressure with time during the discharge and to examine the conditions inside the vessel due to the wave action which follows from the rapidity of the discharge. The vessel is cylindrical in shape and 4 in. long, with the release port at one end, the port being opened at a known rate, as given in Fig. 4. The rate of opening assumed is such that the effective port area increases uniformly from zero to one-half of the area of the vessel in 0·003 second, and thereafter remains constant. The effective area incorporates a coefficient of discharge for the orifice, assumed to have been

obtained experimentally for the particular shape of the port and for the pressure conditions of the problem. The initial condition of the gas inside the vessel is chosen to represent the products of combustion in an engine cylinder, with a pressure of 75 lb. per square inch absolute, and a temperature of 1,200 deg. C. The composition of the gas is such that the initial density is 0·077 lb. per cubic foot, and the mean value of the adiabatic index, γ, over the range of temperature involved, is 1·32.

First Stage of the Discharge. (*First Interval.*) For the conditions chosen, the speed of sound in the gas at rest, given by $\sqrt{\frac{g\,\gamma\,p_0}{\rho_0}}$, is 2,447 ft. per second. The time for a wave to travel twice the length of the vessel is therefore about 0·00027 second. From Fig. 4 the average value of $\frac{a_2}{a_1}$ for this interval is 0·028, giving $\frac{a_1}{a_2} = 35\cdot7$. The actual duration of the interval will be slightly greater than the time calculated on the basis of 2,447 ft. per second, since a lower value, c_0^1, applies during the forward movement of the wave after reflection. This latter value cannot be determined until the end of the calculations for the interval, but in any case the difference between 0·00027 second and the actual time for the interval is small, and does not affect appreciably the mean value of $\frac{a_1}{a_2}$ for the interval.

With the above value of $\frac{a_1}{a_2}$, and with $p_0 = 75 \times 144$ lb. per square foot, equation (8) becomes

$$\{223\cdot1\,(3\cdot082\,p_1^{0\cdot7575} - p_1^{0\cdot8788})\}^{0\cdot2758} + 33\cdot22\,p_1^{0\cdot1212} - 6\cdot25\,p_1^{0\cdot2424} = 51\cdot2,$$

p_1 being expressed in pounds per square foot. Using values of 68, 70, 72 and 74 lb. per square inch for p_1, the left-hand side of this equation was plotted against p_1, and it was found that a value of 51·2 (the right-hand side of the equation) corresponds to $p_1 = 73\cdot5$ lb. per square inch.

The velocity with which the gas approaches the port is then obtained from

$$\left(\frac{73\cdot5}{75}\right)^{0\cdot1212} = 1 - \frac{0\cdot32}{2 \times 2447}\,v_1,$$

which gives $v_1 = 38$ ft. per second. For the pressure in the gas at rest after the reflection of the wave equation (2) takes the form

$$(p_0^1)^{\frac{\gamma-1}{2\gamma}} - p_1^{\frac{\gamma-1}{2\gamma}} = \frac{\gamma-1}{2}\,.\,v_1\sqrt{\frac{p_0^{\frac{1}{\gamma}}}{g\,\gamma\,p_0}},$$

from which p_0^1 is found to be 10,350 lb. per square foot or 71·9 lb. per square inch.

The speed of sound at 71·9 lb. per square inch is 2,434 ft. per second; hence the mean value of c_0 for the complete interval is 2,440 ft. per second. The time for the interval, *i.e.*, the time to traverse a total distance of 8 in., is 0·00027 second.

(*Second Interval.*)—The initial pressure is now 71·9 lb. per square inch ($p_0 = 71\cdot9 \times 144$ lb. per square foot). The duration of the first interval being known, the mean value of $\frac{a_2}{a_1}$ for the second interval is obtained from Fig. 4, assuming approximately the same duration. In this way the value of $\frac{a_1}{a_2}$ for the interval is found to be 14·7. These *quantities are inserted in equation* (8) and the new value of p_1 is determined as before. For this interval the pressure p_1 is found to be 66·9 lb. per square inch, the velocity v_1 is 132 ft. per second, the pressure after reflection is 62·3 lb. per square inch, and the time occupied by the double travel of the wave is 0·000273 second.

(*Intervals* 3, 4 *and* 5.)—The results of the calculations for the subsequent intervals are shown in Table I, together with those for intervals 1 and 2, the pressures being given in pounds per square inch. From this it is seen that at the end of the fifth interval the pressure after reflection has fallen to 26·9 lb. per square inch.

TABLE I.

Interval	1.	2.	3.	4.	5.
p_0, lb. per sq. in.	75·0	71·9	62·3	51·2	39·8
$\frac{a_1}{a_2}$	35·7	14·7	8·4	6·2	4·77
p_1, lb. per sq. in.	73·5	66·9	56·5	45·2	33·6
v_1, ft. per sec.	38	132	177	219	285
p_0^1, lb. per sq. in.	71·9	62·3	51·2	39·8	26·9
Duration, sec. $\times 10^{-3}$	0·270	0·273	0·270	0·287	0·298

It is necessary now to examine the conditions of flow, in order to find the point at which the critical pressure ratio no longer applies to the flow through the port. Since the velocity v_1 appears in the equation for the critical pressure ratio, it is necessary to choose some value of v_1 for insertion in equation (4). If the value determined for the fifth interval (285 ft. per second) be used, the value of p_1 at which the exit pressure is equal to the atmospheric pressure is found from

$$\left(\frac{14\cdot7 \times 144}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma+1} + \frac{\gamma-1}{\gamma+1}\,\frac{285^2\,\rho_0}{2\,g\,\gamma\,p_0^{\frac{1}{\gamma}}\,.\,p_1^{\frac{\gamma-1}{\gamma}}},$$

which gives $p_1 = 3{,}860$ lb. per square foot, or 26·8 lb. per square inch. With a higher value for v_1 the critical value of p_1 would be correspondingly lower. It follows that the end of this stage of the expansion is reached when p_1 falls to 26·8 lb. per square inch, or some slightly lower value if v_1 be greater than 285 ft. per second. For the sixth interval it is clear that, since p_0 is 26·9 lb. per square inch, the value of p_1 will be below the critical pressure, and the conditions of flow are those of the second stage.

(*To be continued.*)

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RAPID DISCHARGE OF GAS FROM A VESSEL INTO THE ATMOSPHERE.

By E. GIFFEN, Ph.D., M.I.Mech.E.

(*Concluded from page 155.*)

Second Stage of the Discharge.—Equation (11), page 154, *ante*, is now used instead of (8), but otherwise the work is similar. In the sixth interval (the first in this stage), the initial pressure is 26·9 lb. per sq. in., $\frac{a_1}{a_2}$ is 3·87, from Fig. 4, page 154, *ante*, and the atmospheric pressure is taken as 14·7 lb. per sq. in. When these are inserted in (11), the equation becomes

$$6\cdot33287 \times 10^{-3}\,p_1^{1\cdot81515} - 4\cdot65316 \times 10^{-3}\,p_1^{1\cdot69636} + 8\cdot54753 \times 10^{-4}\,p_1^{1\cdot75757} + 34\cdot02\,p_1^{\cdot1212} - 7\cdot25\,p_1^{\cdot2424} = 39\cdot94,$$

p_1 again being expressed in lb. per sq. ft. The first three terms of this equation give the difference of nearly equal quantities, and it has been necessary to work with the larger number of significant figures shown above, using seven-figure logarithms, in order to obtain consistent results. With values of 21, 22, 23 and 24 lb. per square inch for p_1, the left-hand side of the equation gives 40·34, 39·98, 39·51 *and* 39·33, *respectively*. From these the actual value of p_1 during the sixth interval is found to be 22·1 lb. per square inch. The velocity v_1 of the gas in the vessel, and the pressure p_0^1 in the gas after the wave is reflected at the closed end, are then found as before, and the work is repeated for successive intervals in the discharge. The results of these calculations are given in Table II,

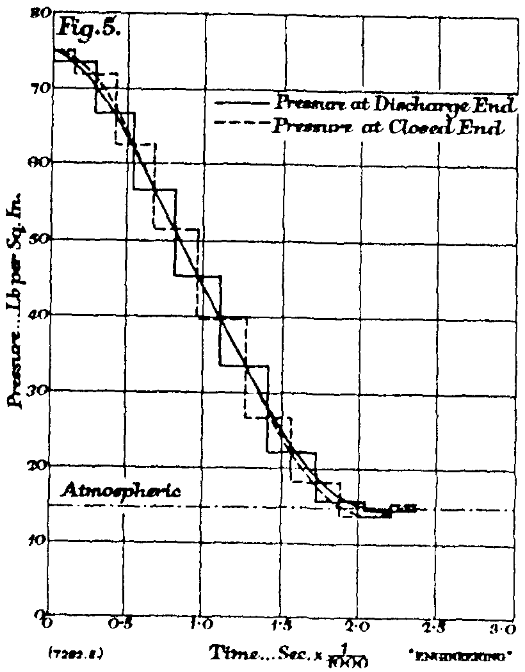


TABLE II.

Interval ..	6.	7.
p_0, lb. per sq. in.	20·9	18·1
$\frac{a_1}{a_2}$	3·87	3·25
p_1, lb. per sq. in.	22·1	15·85
v_1, ft. per sec.	318	205
p_0^1, lb. per sq. in.	18·1	13·85
Duration, second × 10^{-3}	0·320	0·330

Pressures given in lb. per sq. in. abs.

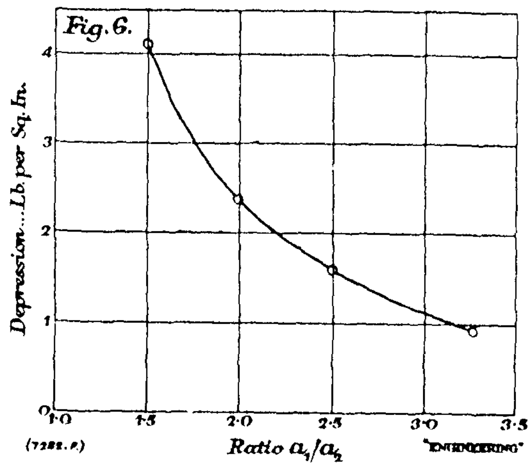
which shows that at the end of the seventh interval the pressure remaining in the vessel, after the forward movement of the reflected wave, is less than that of the atmosphere. Hence the discharge ceases when this wave reaches the port. At the instant at which this last reflected wave reaches the port end of the vessel, the gas is at rest and the pressure is less than that of the atmosphere; the direction of flow will, therefore, be reversed. A positive wave is now generated inside the vessel, the pressure and velocity being determined from the modified equations for the new conditions of flow. This wave is reflected at the closed end and returns to the port, raising further the pressure inside the vessel. The subsequent variations in pressure may, therefore, be calculated by the methods already described, and this has been carried out for one interval. For the equation representing the flow through the port, some assumption must be made regarding the density of the *gas immediately outside the port*. *It has* been taken, for simplicity, that this gas is that discharged from the vessel, and that its density corresponds to adiabatic expansion from the initial conditions. The velocity of approach of the gas in the atmosphere may be taken as zero; the pressure outside the port is p_a, and the pressure inside is p_1; the velocity through the port is then given by

$$v_2^2 = 2\,g\,\frac{\gamma}{\gamma - 1}\,\frac{p_a}{\rho_a}\left\{1 - \left(\frac{p_1}{p_a}\right)^{\frac{\gamma-1}{\gamma}}\right\}.$$

For the forward wave in the vessel equation (2), page 135, *ante*, becomes

$$\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{2\gamma}} = 1 + \frac{\gamma - 1}{2}\cdot\frac{v_1}{c_0},$$

and equation (5) remains as before. With these equations, and continuing from the end of the



seventh interval, it is found that the value of p_1 in the next interval rises to 14·5 lb. per square inch, the pressure after reflection becomes 15·3 lb. per square inch, and the time for the interval is 0·00033 second.

The variation of pressure inside the vessel, during the whole of the discharge period, is shown graphically in Fig. 5, on a base of time. The full lines represent the pressure at the discharge end, p_1 being plotted in the stepped form given by the calculations. The continuous line, drawn through the mid-point of each constant-pressure interval, gives the probable pressure curve when the continuous variation of the port area is taken into account. The variation in pressure at the closed end of the vessel is obtained by plotting p_0^1, changes in pressure at this end lagging half an interval behind those at the port end. This is shown by the broken lines, the continuous curve again representing the variation in pressure with a continuously varying area of the discharge port. From a comparison of the two sets of results, it is seen that the pressure at the closed end is higher than that at the open end at the beginning of the discharge, but lower at the end. Also, a depression exists inside the vessel at the end of the discharge.

The Depression Inside the Cylinder.—For the conditions of the present problem a depression of 0·85 lb. per square inch is found to exist at the closed end of the vessel for one complete interval. With the step-by-step method of calculation employed, the magnitude of the depression depends only upon the conditions existing during the particular interval in which the pressure drops below atmospheric. The important conditions are the pressure p_0 at the beginning of the interval, and the mean area of the port during the interval. Both of these quantities, however, depend upon the whole course of the discharge process, and variations in either will have a considerable effect upon the calculated magnitude of the depression inside the vessel. It is interesting, therefore, to consider the effect of various values of the initial pressure p_0 for the seventh interval, and to find the maximum value of the depression. It is assumed that, owing to variations in the rate of opening of the port, the earlier steps in the discharge are such that the initial pressure for the seventh interval has the various values given in the first line of Table III. the area of the port, however, has the same value as before, *i.e.*, $\frac{a_1}{a_2} = 3·25$ in each case. The corresponding values of p_1 and p_0^1 have been determined, and are shown in the second and third lines of the table.

TABLE III.—*Pressures given in lb. per square inch abs.*

p_0	16·5	17	17·5	18·1	20
p_1	15·15	15·35	15·55	15·85	16·85
p_0^1	14·02	13·79	13·79	13·85	14·14

From these results it is seen that the minimum value of p_0^1 is about 13·8 lb. per square inch, or the maximum depression is 0·9 lb. per square inch, and this occurs when p_0 is 17 lb. to 17·5 lb. per square inch. With a higher value of p_0, the intensity of the pressure wave is correspondingly greater, but p_1 is still so much higher than the atmospheric pressure that, after reflection, the depression becomes smaller. With a lower value of p_0, the intensity of the pressure wave is so much reduced that, although p_1 also decreases, the depression after reflection is reduced. The other important condition

affecting the depression in the vessel is the area of the port during the last discharge interval. The effect of this variable has been investigated for values of $\frac{a_1}{a_2}$ of 2·5, 2·0 and 1·5. In each case a series of values of p_0 is chosen, and p_1 and p_0^1 are determined as before. The results of this work are collected in Table IV, and show that the mini-

TABLE IV.—*Pressures given in lb. per square inch abs.*

$\frac{a_1}{a_2} = 2{\cdot}5$	p_0	19	20	21	22	23
	p_1	15·85	16·25	16·75	17·25	17·8
	p_0^1	13·16	13·13	13·24	13·42	13·66
$\frac{a_1}{a_2} = 2{\cdot}0$	p_0	20	22	23	25	
	p_1	15·82	16·55	16·95	17·90	
	p_0^1	12·44	12·32	12·35	12·59	
$\frac{a_1}{a_2} = 1{\cdot}5$	p_0	22	24	25	27	
	p_1	15·65	16·18	16·50	17·08	
	p_0^1	10·97	10·69	10·62	10·84	

mum pressure in the vessel, after reflection of the wave, becomes progressively less as the area of the port is increased, and also that the minimum pressures are obtained with increasing values of the pressure p_0 at the beginning of the interval. In order to show better the effect of a larger port area upon the depression in the vessel, the maximum depression below the atmospheric pressure is plotted against the ratio $\frac{a_1}{a_2}$ in Fig. 6. The maximum depression, as calculated above, increases from 0·9 lb. per square inch when $\frac{a_1}{a_2}$ is 3·25, to 4·1 lb. per square inch when the area ratio is 1·5, the increase becoming progressively greater as the size of the port is increased.

The conclusions to be drawn from the results shown in Table IV and in Fig. 6 require some modification when the continuous variation of the port area is taken into account. In this case the pressure inside the vessel varies continuously, and not in a series of steps; so that the propagation of waves at one end, and their reflection at the other, is going on continuously. From the method of calculation used above, with a single intense pressure wave generated at the beginning of each complete interval, the magnitude of the depression is found to vary from zero to a maximum according to the value chosen for the initial pressure at the beginning of the interval. With continuously varying pressure in the cylinder, however, the instant at which any particular interval is assumed to begin is entirely fortuitous; also the changes that occur, both in the area of the port and in the pressure, are not, in reality, concentrated at that particular instant, but are distributed over the whole interval. The action is therefore that of a very large number of waves, each of very small magnitude, closely following each other. In the present problem the time for the double movement of a wave through the vessel is about 0·00033 second, when the pressure is in the neighbourhood of atmospheric. Only those waves which leave the port end of the vessel earlier than 0·00033 second before the end of the discharge can arrive back at the port before the pressure there falls to that of the atmosphere. Hence, all the waves that leave during the last 0·00033 second contribute cumulatively to the depression produced. With continuously varying port area, therefore, the critical effect of the pressure at some particular instant does not apply, and it is the general slope of the pressure curve near the end of the discharge that determines the magnitude of the depression. This slope of the pressure curve follows both from the shape of the pressure curve during the earlier stages of the discharge and from the area of the port during the later stages. For the same reasons, also, the magnitude of the depression in the vessel would be somewhat smaller than the maximum value calculated above.

The general treatment developed above may be applied to any problem of rapid discharge from a cylindrical pressure vessel for which the rate of variation of the port area is known. The subdivision of the discharge period into a number of intervals, during each of which the port area is assumed to remain constant, involves a certain degree of approximation, and the accuracy of the mean pressure curve derived from the calculations depends upon the number of intervals chosen. In the problem to which the method has been applied, it was convenient to choose each interval as equal to the time required for the double movement of a wave through the contents of the vessel; in other cases it might be necessary, for accurate results, to choose some sub-multiple of this as the time interval. With the step-by-step method of calculation it is then possible to trace the variation in pressure at any point in the vessel during the whole process of discharge, and to determine the time required for the pressure to fall to that of the atmosphere. Further, as a result of the intense wave action that accompanies the rapid discharge, there is, in general, a depression in the vessel at the end of the discharge period, and the calculation gives an estimate of the magnitude of this depression.

Part II. *Based on Assumption of Uniform Pressure in the Vessel.*—It is both interesting and instructive to compare the results obtained above with those given by a method of calculation in which the wave action inside the vessel is neglected. In the latter case, the problem is simplified by the assumption that the pressure is uniform throughout the vessel; also, as a result of this assumption, a further simplification is that the velocity of the gas inside the vessel is so small compared with the other velocities involved that it may be taken as zero. The latter assumption permits the use of the simpler expression for the critical ratio between the pressures inside the vessel and at exit from the discharge port. If p be taken to represent the pressure inside the vessel, assumed to be uniform throughout the vessel, and the other symbols are as before, then equation (4), page 136, *ante*, takes the form

$$\left(\frac{p_2}{p}\right)^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma+1}. \quad . \quad . \quad (12)$$

In equation (3), for the velocity through the port, v_1 becomes zero, so that

$$v_2^2 = 2g\frac{\gamma}{\gamma-1}\frac{p}{\rho}\left\{1-\left(\frac{p_2}{p}\right)^{\frac{\gamma-1}{\gamma}}\right\} \quad . \quad (13)$$

The discharge period is divided into two stages, as before. In the first, p_2 is greater than the atmospheric pressure p_a, and is related to p by equation (12); also, v_2 is the speed of sound at pressure p_2. In the second stage, p_2 is equal to p_a throughout, and v_2 is less than the speed of sound at pressure p_a.

First Stage.—If the weight of gas in the vessel at any instant be denoted by w, its density by ρ, and the volume of the vessel by V, then

$$w = \rho\,\mathrm{V}.$$

When ρ is expressed in terms of p, as before, this equation becomes

$$w = \frac{\mathrm{V}\,\rho_0}{p_0^{\frac{1}{\gamma}}}\cdot p^{\frac{1}{\gamma}},$$

where p_0 and ρ_0 refer to the initial condition at the beginning of the discharge. This is differentiated with respect to time, t, to obtain the rate of change of the weight of gas in the vessel, giving

$$\frac{dw}{dt} = \frac{\mathrm{V}\,\rho_0}{\gamma\,p_0^{\frac{1}{\gamma}}}\cdot p^{\frac{1-\gamma}{\gamma}}\frac{dp}{dt} \quad (14)$$

But the rate of change of the quantity of gas in the vessel is the same as the rate of flow through the discharge port, that is,

$$\frac{dw}{dt} = -\rho_2\,v_2\,a_2, \quad . \quad . \quad (15)$$

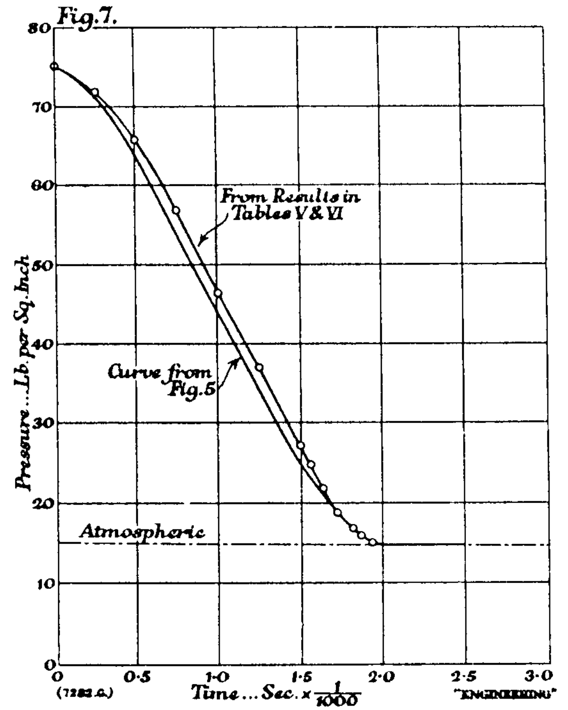
the negative sign being used because the quantity of gas in the vessel is decreasing. The density ρ_2 may be expressed in terms of p, since

$$\rho_2 = \rho\left(\frac{p_2}{p}\right)^{\frac{1}{\gamma}},$$

and, substituting for ρ in terms of p, and for $\frac{p_2}{p}$ from equation (12), this gives

$$\rho_2 = \frac{\rho_0}{p_0^{\frac{1}{\gamma}}}\cdot p^{\frac{1}{\gamma}}\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}.$$

Fig. 7.



In order to express v_2 in terms of p, $\frac{p_2}{p}$ in equation (13) may be replaced by

$$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$

TABLE V.—*1st Stage.*

t, sec. × 10^{-3}	0·25	0·5	0·75	1·0	1·25	1·5
p, lb. per sq. in. abs.	71·9	65·9	56·8	46·7	37·0	27·2

and the equation reduces to

$$v_2 = \sqrt{\frac{2\,g\,\gamma}{\gamma+1}\cdot\frac{p_0^{\frac{1}{\gamma}}}{\rho_0}}\cdot p^{\frac{\gamma-1}{2\gamma}}.$$

These expressions for ρ_2 and v_2 are substituted in (15), and the two values of $\frac{d\,w}{d\,t}$ in (14) and (15) may then be equated, giving

$$p^{\frac{1-3\gamma}{2\gamma}}\,d\,p = -\,a_2\,\frac{\gamma}{V}\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\sqrt{\frac{2\,g\,\gamma}{\gamma+1}\cdot\frac{p_0^{\frac{1}{\gamma}}}{\rho_0}}\cdot d\,t \quad . \quad . \quad (16)$$

The solution of this equation gives the rate of variation of the pressure inside the cylinder, but first it is necessary to be able to express the port area a_2 in terms of t.

TABLE VI.—2ND STAGE.

t, beginning of interval, sec. × 10^{-3}	1·5	1·5541	1·6328	1·7283	1·8100	1·8639
p_A, lb. per sq. in. abs. . .	27·2	25·0	22·0	19·0	17·0	16·0
δp, lb. per sq. in. abs. . .	2·2	3·0	3·0	2·0	1·0	1·0
p, lb. per sq. in. abs. . .	26·1	23·5	20·5	18·0	16·5	15·5
a_2, sq. ft. . .	0·0117	0·0120	0·0124	0·0127	0·0130	0·0133
δt, sec. × 10^{-3} . .	0·0541	0·0787	0·0956	0·0817	0·0539	0·0798

In the problem to which the result is to be applied the area of the discharge port increases uniformly with time, that is $a_2 = c\,t$, where c is a constant. This may be inserted in (16) and both sides of the equation may now be integrated, leading to

$$p = \frac{1}{\left\{\left(\frac{1}{p_0}\right)^{\frac{\gamma-1}{2\gamma}} + \frac{c\,(\gamma-1)}{4\,V}\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\sqrt{\frac{2\,g\,\gamma}{\gamma+1}\cdot\frac{p_0^{\frac{1}{\gamma}}}{\rho_0}}\cdot t^2\right\}^{\frac{2\gamma}{\gamma-1}}}. \quad (17)$$

Second Stage.—The beginning of this stage is reached when the pressure inside the vessel falls to a value such that

$$\left(\frac{p_a}{p}\right)^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma+1}.$$

Thereafter the pressure at exit from the port is constant and equal to p_a, and the pressure in the vessel falls continuously until it reaches this value. At the beginning of this period the exit velocity is equal to that of sound at a pressure p_a, and decreases continuously to zero as the internal pressure approaches p_a. An expression for the exit velocity v_2 may be obtained from (13), making the same substitution as before for ρ, and substituting p_a for p_2. This gives

$$v_2 = \sqrt{\frac{2\,g\,\gamma}{\gamma-1}\cdot\frac{p_0^{\frac{1}{\gamma}}}{\rho_0}\cdot p^{\frac{\gamma-1}{2\gamma}}\left\{1-\left(\frac{p_a}{p}\right)^{\frac{\gamma-1}{\gamma}}\right\}^{\frac{1}{2}}}.$$

The density to be used in equation (15) is now ρ_a, and for this may be substituted

$$\frac{\rho_0}{p_0^{\frac{1}{\gamma}}}\cdot p_a^{\frac{1}{\gamma}}.$$

These expressions for v_2 and ρ_a are substituted in (15), and the value of $\frac{d\,w}{d\,t}$ equated with that of (14), giving

$$\frac{d\,p}{p^{\frac{3(\gamma-1)}{2\gamma}}\left\{1-\left(\frac{p_a}{p}\right)^{\frac{\gamma-1}{\gamma}}\right\}^{\frac{1}{2}}} = -\,a_2\,\frac{\gamma}{V}\,p_a^{\frac{1}{\gamma}}\sqrt{\frac{2\,g\,\gamma}{\gamma-1}\cdot\frac{p_0^{\frac{1}{\gamma}}}{\rho_0}}\cdot d\,t \quad . \quad (18)$$

The simplest method of dealing with this equation, in the case of a numerical example, is to use a step-by-step process of integration. The time for this stage of the discharge is divided up into small finite intervals. During any interval, t_A to t_B, corresponding to $\delta\,t$, the pressure p changes from p_A to p_B; this is $\delta\,p$. The value of p_A for the interval is known from the calculation for the previous interval; the average value of p for the interval is $\frac{1}{2}(p_A + p_B)$; and the value of a_2 is the average value for the interval. For any chosen value of $\delta\,p$ the corresponding value of $\delta\,t$ follows directly.

Application to Typical Problem. The results obtained above are now applied to the practical problem dealt with earlier, in order to determine the variation of the pressure inside the vessel with time, during the discharge period. The diameter of the cylindrical vessel was 2·56 in., so that the initial volume V was 0·0119 cub. ft. The corresponding rate of opening of the port was given by $a_2 = 5\cdot96\,t$, where a_2 is in sq. ft. and t in seconds. Equation (17) then reduces to

$$p = \frac{1}{\{0\cdot325 + 18{,}570\,t^2\}^{8\cdot26}}$$

and this is used with various values of t until the pressure p has fallen to the value given by the critical pressure ratio. In this case, $\frac{p_a}{p} = 0\cdot541$, so that the end of the first stage of the discharge is reached when p falls to $\frac{14\cdot7}{0\cdot541}$, or 27·2 lb. per square inch. This is reached at 0·0015 second after the beginning of discharge. For the second stage, equation (18) is employed and the calculation is carried out in a series of steps. The negative sign in the equation merely indicates a decreasing pressure, and may be disregarded if this is borne in mind. With the units in lb., feet and seconds, the equation then takes the form

$$\delta\,t = \frac{\delta\,p}{72\cdot66\times10^{6}\,a_2\,p^{0\cdot3637}\left\{1-\left(\frac{p_a}{p}\right)^{0\cdot2424}\right\}^{\frac{1}{2}}}.$$

In the first interval of this stage the initial pressure, p_A, is 27·2 lb. per square inch. The value chosen for $\delta\,p$ is 2·2 lb. per square inch, so that p_B is 25 lb.

per square inch. The pressure p for the interval is 26·1 lb. per square inch, and the area of the discharge port is, from $a_2 = 5 \cdot 96\, t$, about 0·0117 sq. ft. When these are inserted in the above equation, the value of δt is found to be 0·000054 second. This gives the time when the pressure has fallen to 25 lb. per square inch, and this is then the initial pressure for the next interval.

Tables V and VI, page 182, show the results of a series of calculations for both stages of the discharge, and the variation of pressure is plotted against time in Fig. 7. On the same figure is shown, for comparison, the curve obtained previously in Fig. 5, for the pressure at the discharge port. As is seen from the curves, there is, in this case, but little difference between the results obtained by the two methods of calculation, so that the latter method, which is much simpler to use, may be regarded as giving a good idea of the actual time of discharge, and of the rate of decrease of pressure inside the vessel. A very important difference between the results of the two methods of calculation, however, is that with the latter there is no indication of a pressure gradient inside the vessel; indeed, this is implicit in the assumption on which the method is based. It follows, also, that there can be no indication of a depression inside the vessel at the end of the discharge period. It is on this point that the first method of calculation represents an advance on the second, as approaching nearer to the true conditions inside the vessel, and taking into account the mechanism whereby the pressure changes are considered to be transmitted throughout the vessel. It is, in fact, mainly in order to demonstrate that a depression is to be expected when a pressure vessel discharges rapidly into the atmosphere that the method of analysis in the first part of this article has been developed. The additional numerical work involved in the application of that method would only be justified when it is required to obtain an estimate of the magnitude of the depression.

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A THEORY OF THE KADENACY SYSTEM.

By E. W. GEYER, B.Sc., Ph.D.

A CONSIDERABLE amount of discussion regarding the Kadenacy system of exhausting gases from an internal-combustion engine cylinder has appeared in the correspondence columns of ENGINEERING, but it is somewhat surprising that no effort has been made by any of the investigators to place the matter on a rational basis.* It is hoped, therefore, that the following treatment may prove of general interest.

Consider the vessel A E F D, Fig. 1, page 464, having the internal capacity V cub. ft. and containing W lb. of gas at the absolute pressure p_1 and absolute temperature T_1. If the cover on the nozzle face G H is suddenly removed, the gas pressure in the vessel rapidly drops to that of the space beyond the nozzle and will, under certain conditions, drop below this value. At the instant when the pressure in the vessel is equal to the external pressure p_b let the mass of gas in the vessel be W_1 lb. and let the mass of gas which has escaped from the vessel be W_2 lb., *i.e.*, $W_1 + W_2 = W$. If the space A B C D contained the mass W_1 lb. of gas before discharge occurred, then the space B E F C contained W_2 lb. of gas and at the instant of equalisation of pressure the face B C will have arrived at the nozzle entrance. Let its approach velocity there be V_1 ft. per second. Since the velocity of the gas at A D is zero, the kinetic energy of the gas in the vessel is $\frac{1}{3}\frac{W_1 V_1^2}{2g}$. This is based on the assumption of a uniform increase in velocity from one end of the cylinder to the other.

If the cross-sectional areas of the vessel and nozzle are A_1 and A_2, the velocity of the gases leaving the nozzle is given by $V_2 = \frac{V_1 A_1}{A_2}$. This follows since the pressure at the nozzle exit and the pressure in the vessel are equal and hence the specific volumes in these regions are equal. The gases which have left the vessel may be assumed to occupy the imaginary cylinder shown in dotted outline as an extension on the right of the vessel. These gases have pushed the imaginary piston G H, on the outer side of which the constant pressure p_b acts, to the position K L, and at this instant the static pressure within the vessel equals p_b. The decrease in internal energy is given by $J\, W\, c_v (T_1 - T_2)$, where $T_2 = T_1 \left(\frac{p_b}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$. This assumes isentropic expansion throughout. The work which has been performed so far against the external pressure is $144\, p_b\, W_2\, v_2$, where v_2 is the specific volume of the gas after expansion to p_b and is given by the relationship $v_2 = \frac{R\, T_2}{144\, p_b}$. The kinetic energy of the gas which has escaped from the cylinder is given, to a first approximation, by

$$\frac{W_2 V_2^2}{2g} = \frac{W_2}{2g}\left(\frac{A_1 V_1}{A_2}\right)^2$$

so that the total kinetic energy given to the gas is

$$\frac{1}{3}\frac{W_1 V_1^2}{2g} + \frac{W_2}{2g}\left(\frac{A_1 V_1}{A_2}\right)^2.$$

We thus have

$$J\, W\, c_v (T_1 - T_2) = 144\, p_b\, v_2\, W_2 + \frac{V_1^2}{2g}\left\{\frac{1}{3} W_1 + W_2 \left(\frac{A_1}{A_2}\right)^2\right\} \quad . \quad . \quad (1)$$

From this expression, the value of V_1 can be calculated. Due to the kinetic energy of the gases in the vessel and the additional kinetic energy of the mass of gas which occupies the nozzle or any actual extension of the vessel, further work can be performed against the external pressure p_b. If W_3 be the weight of gas contained in the nozzle or in any extension, the work which can be performed against p_b is

$$\frac{1}{3}\frac{W_1 V_1^2}{2g} + \frac{W_3 V_2^2}{2g} \quad . \quad . \quad . \quad (2)$$

and this equals $144\, p_b\, V$, where V is the increase in volume of the gas beyond the face G H. On the assumption that this expansion is adiabatic, we have the relationship

$$p_b (V')^\gamma = p\,(V' + V)^\gamma \quad . \quad . \quad (3)$$

where V' is the volume of the vessel and the nozzle. The value $p_b - p$ is the attainable depression in the vessel under ideal conditions. The effect of a cylindrical extension, such as an exhaust pipe, to the vessel is to increase this depression as shown by the second term of the expression given above for the additional work performed against p_b, *i.e.*, by the expression $\frac{W_3 V_2^2}{2g}$.

From the data provided by the tests of Professor Davies, as described in the issue of ENGINEERING for January 5, 1940, page 17, the depression for the case in which the initial pressure is 60 lb. per square inch is found by means of the above treatment as follows: Assuming that the initial temperature in the vessel is 65 deg. F., and that the atmospheric pressure is 14·7 lb. per square inch, the initial absolute values of the temperature and pressure are $T_1 = 525$ deg. F. abs. and $p_1 = 14 \cdot 7 + 60 = 74 \cdot 7$ lb. per square inch abs. The temperature after expansion to $p_b = 14 \cdot 7$ lb. per square inch abs. is $T_2 = T_1 \left(\frac{p_b}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 525 \div \left(\frac{74 \cdot 7}{14 \cdot 7}\right)^{\frac{0 \cdot 4}{1 \cdot 4}} = 330$ deg. F. abs. The initial weight of air in the vessel is $W = 144\, \frac{p_1 V}{R\, T_1} =$

* This article was received at about the same time as that of Dr. E. Giffen, which was published in our 150th volume (1940), on pages 134, *et seq.*, but the publication of Dr. Geyer's article was delayed owing to circumstances arising out of the war.—ED., E.

$144 \times 74 \cdot 7 \times \dfrac{0 \cdot 0945}{53 \cdot 3} \times 330 = 0 \cdot 0363$ lb. The weight of air in the vessel when the pressure drops to $14 \cdot 7$ lb. per square inch abs. is $W_1 = 144 \times 14 \cdot 7 \times \dfrac{0 \cdot 0945}{53 \cdot 3} \times 330 = 0 \cdot 0114$ lb., and at this instant the specific volume is $v_2 = \dfrac{R\,T_2}{144\,p_b} = \dfrac{53 \cdot 3 \times 330}{144 \times 14 \cdot 7} = 8 \cdot 33$ cub. ft. per lb. The weight of air discharged up to this instant is $W_2 = W - W_1 = 0 \cdot 0363 - 0 \cdot 0114 = 0 \cdot 0249$ lb. The ratio $\dfrac{A_1}{A_2} = \left(\dfrac{4}{3 \cdot 625}\right)^2 = 1 \cdot 22$, so that $\left(\dfrac{A_1}{A_2}\right)^2 = 1 \cdot 49$. Equation (1) thus gives

$$\frac{V_1^2}{2g} = [778 \times 0 \cdot 0363 \times 0 \cdot 169\ (525 - 330)$$
$$- 144 \times 14 \cdot 7 \times 8 \cdot 33 \times 0 \cdot 0249]$$
$$\div (\tfrac{1}{2} \times 0 \cdot 0114 + 0 \cdot 0249 \times 1 \cdot 49)$$
$$= \frac{930 - 439}{0 \cdot 0408} = \frac{491}{0 \cdot 0408} = 12{,}050 \text{ ft.-lb.}$$

In the vessel used by Professor Davies the length of the parallel portion at outlet is $\frac{3}{4}$ in. and the weight of air contained in this, when the pressure becomes $14 \cdot 7$ lb. per square inch abs., is

$$W_3 = \frac{\pi \times 3 \cdot 6252 \times 0 \cdot 75}{4 \times 1{,}728 \times 8 \cdot 33} = 0 \cdot 000537 \text{ lb.}$$

Hence from equation (2) the available energy for the performance of work against p_b is $12{,}050\ (0 \cdot 0038 + 1 \cdot 49 \times 0 \cdot 000537) = 52 \cdot 2$ ft.-lb. The increase in volume of the gas beyond the nozzle exit is, therefore, $V = \dfrac{52 \cdot 2}{144 \times 14 \cdot 7} = 0 \cdot 0247$ cub. ft. Equation (3) gives $p = \dfrac{14 \cdot 7}{(0 \cdot 1192/0 \cdot 0945)^{1 \cdot 4}} = 10 \cdot 6$ lb. per square inch abs., so that the depression in the vessel is $14 \cdot 7 - 10 \cdot 6 = 4 \cdot 1$ lb. per square inch abs. Without the $\frac{3}{4}$-in. cylindrical extension the available energy is $12{,}050 \times 0 \cdot 0038 = 45 \cdot 7$ ft.-lb. and $V = \dfrac{45 \cdot 7}{144 \times 14 \cdot 7} = 0 \cdot 0216$ cub. ft. This gives $p = 14 \cdot 7 \div \left(\dfrac{0 \cdot 1161}{0 \cdot 0945}\right)^{1 \cdot 4} = 11$ lb. per square inch abs., so that the depression in the vessel is $14 \cdot 7 - 11 \cdot 0 = 3 \cdot 7$ lb. per square inch. Similar calculations have been carried out for initial pressures of 10, 20, 30, 40 and 50 lb. per square inch, and the results are shown in Table I.

Fig. 1.

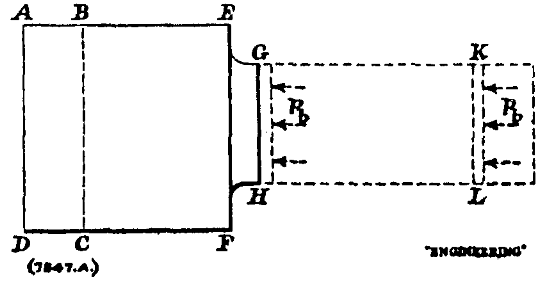


Fig. 2.

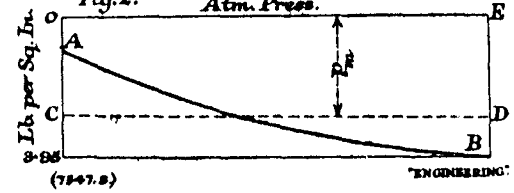
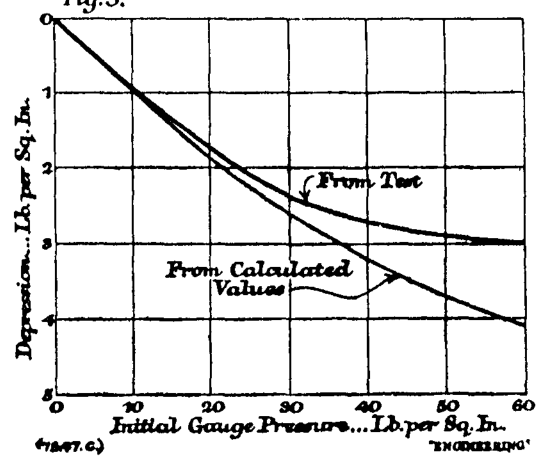


Fig. 3.



Professor Davies shows from his test results that the depression is not uniform throughout the vessel. Let the variation in pressure be represented by the curve A B in Fig. 2. If the vessel could be closed suddenly the resulting uniform depression which would be attained in the vessel would be represented by O C where the area O C D E is equal to the area O A B E. From the curve given by Professor Davies in his article it is found that the mean uniform depression is $0 \cdot 69$ of the maximum depression. His maximum depressions, given in Fig. 4 of his article, have therefore been multiplied by this factor in order that a comparison may be made between his measured values and the values obtained by calculation. These are given in Table I, below. The calculated results, along with those given by Professor Davies from his tests, have been plotted in Fig. 3.

It will be seen from equation (1) that when, as is the case in practice, the outlet area is considerably less than the cross sectional area of the vessel, the available energy causing the depression is considerably reduced. Thus, in the example worked out in detail above, if $\dfrac{A_2}{A_1}$ is taken as $\dfrac{1}{6}$ in place of $\dfrac{1}{1 \cdot 22}$, equation (1) gives

$$\frac{V_1^2}{2g} = [778 \times 0 \cdot 0363 \times 0 \cdot 169\ (525 - 330)$$
$$- 144 \times 14 \cdot 7 \times 8 \cdot 33 \times 0 \cdot 0249]$$
$$\div (\tfrac{1}{2} \times 0 \cdot 0114 + 0 \cdot 0249 \times 36)$$
$$= 546 \text{ ft.-lb.}$$

The available kinetic energy from the gas in the vessel is thus $546 \times 0 \cdot 0038 = 2 \cdot 08$ ft.-lb., so that $V = \dfrac{2 \cdot 08}{144 \times 14 \cdot 7} = 0 \cdot 000983$ cub. ft. This gives $p = 14 \cdot 7 \div \left(\dfrac{0 \cdot 0955}{0 \cdot 0945}\right)^{1 \cdot 4} = \dfrac{14 \cdot 7}{1 \cdot 014} = 14 \cdot 5$ lb. per square inch abs. The depression is thus only $0 \cdot 2$ lb. per square inch. It seems clear that unless the area at outlet is a considerable fraction of the cross-sectional area of the vessel the depressions obtained are negligibly small. The implication is, therefore, that unless care is taken to provide a relatively large discharge area, a system depending on such depressions is likely to be inoperative.

TABLE I.

Initial Gauge Pressures, lb. per sq. in.	10	20	30	40	50	60
Depressions in lb. per sq. in. — Without extension	0·8	1·5	2·3	2·8	3·3	3·7
Depressions in lb. per sq. in. — With $\frac{3}{4}$ in. extension	1·0	1·8	2·6	3·2	3·7	4·1
Adjusted depressions from Professor Davies' tests ..	0·99	1·69	2·42	2·72	2·90	3·04

ENGINEERING LETTERS TO THE EDITOR

7-18-41 p.54

A THEORY OF THE KADENACY SYSTEM.

To the Editor of Engineering.

Sir,—In an article published in your issue of June 13, on page 463, Dr. E. W. Geyer put forward a theory of the Kadenacy system. I should be grateful if you would allow me to comment on his treatment of the problem. The theory, which is based on the striking of an energy balance at the instant when the pressure in the vessel has fallen to atmospheric, appears to be open to criticism owing to the treatment of one of the kinetic energy items. Dr. Geyer states that, to a first approximation, the kinetic energy of the gas which has left the vessel up to the instant of pressure equalisation is $\frac{W_2 V_2^2}{2g}$, where W_2 is the weight of the escaped gas and V_2 is the velocity in the port at this instant. I suggest, however, that this statement is incorrect and that the total kinetic energy of the escaped gas will, in fact, considerably exceed the value assumed by Dr. Geyer, since consideration shows that the velocity of the emergent gases will vary during the discharge and will, in general, be higher during the interval prior to the instant of pressure equalisation than at the instant itself. This has indeed been demonstrated by the sound and logical arguments advanced by Dr. Giffen in an article published in your issues of August 16 and 23 and September 6, 1940, which, as your footnote indicates, Dr. Geyer had no opportunity of seeing before his own article was written.

This under-estimation of the kinetic energy of the escaped gases may possibly be responsible for the appreciable discrepancy between calculated and experimental results shown in Fig. 3 of Dr. Geyer's article, and I suggest that its effect is to make the results obtained by this method of calculation of questionable value. It is also worthy of note that the treatment adopted by Dr. Geyer is confined to a determination of the maximum value of the depression produced. Since the chief interest in the problem lies in the scavenge effect which can be produced in an actual engine, it appears that no treatment can be satisfactory unless it allows not only the magnitude of the depression, but also the duration and the instant of its occurrence to be obtained. It is also interesting to note that neither of the theories referred to above indicates the existence of the ultra-high velocities to which reference has been made by supporters of the Kadenacy claims nor throws any light on the mystical and unexplained term "ballistic discharge."

Your faithfully,
F. K. Bannister.

The University, Birmingham.
July 5, 1941.

8-41 p.94

Sir,—In reply to Mr. Bannister's criticism of my article bearing the above title, it should be observed that since I equate the total energy in the system at one instant to the total energy in the same system at some later instant no consideration need be given to changes which may occur at any intermediate instant. It may be, however, that Mr. Bannister really means that he does not agree with me when I assume that the whole mass of escaped gas is moving with the velocity V_2, although he does not clearly state this. If this is so I would point out that the highest velocity, under controlled conditions, of the escaped gases occurs when the valve is first opened, since the pressure difference between the inside and outside of the vessel has then its greatest value. This velocity is, of course, easily determined. In the example which I quoted it is given by $223 \cdot 8\sqrt{c_p\ (T_1 - T_2)} = 1{,}530$ ft. per second. Actually, under uncontrolled conditions, the emergent velocity would be the acoustic velocity at the critical pressure of $0 \cdot 528 \times 74 \cdot 7 = 39 \cdot 4$ lb. per square inch absolute and temperature of

$$525 \left(\frac{39 \cdot 4}{74}\right)^{\frac{0 \cdot 4}{1 \cdot 4}} = 437 \text{ deg. F. abs.}$$

The velocity is then $V_2 = 223 \cdot 8\sqrt{0 \cdot 24(525 - 437)} =$ 1,025 ft. per second. The value of V_2 which is obtained in my treatment is 1,060 ft. per second, so that I feel that I am still justified in accepting this as a first approximation, especially when the more probable figure for the initial velocity is that given by uncontrolled conditions, namely, of the order of 1,025 ft. per second.

I agree with Mr. Bannister that it would be more satisfactory if the treatment gave the duration of the depression and instant of its occurrence. I have, however, already dealt with the time of discharge from closed vessels in an article which appeared in *The Engineer* on July 21, 1939. From the treatment given there it is possible to determine the time taken for the pressure to drop to any assumed value. My main reason for submitting the article on the Kadenacy system was that at the time it was written there was considerable controversy regarding the existence of depressions of the magnitude claimed by the adherents of the system.

Yours faithfully,
E. W. Geyer.

James Watt Engineering Laboratories,
The University, Glasgow.
July 25, 1941.

8-15-41 p.134

Sir,—In his letter of July 25, published in your issue of August 1, on page 94, Dr. Geyer correctly interprets my criticism of his article on the Theory of the Kadenacy System. I could not and, in spite of the figures he gives, still cannot agree that the escaped gases are moving with constant velocity. I do agree with him that, in the first part of the discharge, the velocity in the port is equal to the acoustic velocity under the pressure and temperature conditions prevailing there and accept his figure of 1,025 ft. per second for the initial port velocity. It would appear, however, that this equality between velocity of discharge and acoustic velocity should hold until the cylinder pressure is equal to atmospheric pressure $\div$ 0·528, *i.e.*, until the port pressure has fallen to atmospheric. I submit a calculation of the acoustic velocity in the port for this instant:

$$\text{Port temperature} = 525 \times \left(\frac{14 \cdot 7}{74 \cdot 7}\right)^{\frac{2}{7}} = 330 \text{ deg. F. abs.,}$$

$$\text{density in port} = \frac{144 \times 14 \cdot 7}{53 \cdot 3 \times 330}.$$

Substituting these in the well known formula for acoustic velocity,

$$V = \sqrt{\left(\frac{P g \gamma}{\rho}\right)}$$

gives

$$\sqrt{\left(\frac{14 \cdot 7 \times 144 \times 32 \cdot 2 \times 1 \cdot 4 \times 53 \cdot 3 \times 330}{144 \times 14 \cdot 7}\right)} = 892 \text{ feet per second}$$

as the acoustic velocity, and also as the velocity of discharge in the port at this instant.

I feel that over this part of the discharge alone there is a sufficient variation to render invalid any assumption of uniform velocity, particularly as the square of the velocity appears in the energy equation. It must also be borne in mind that, during the period considered above, the gas is emerging at a pressure higher than that of the atmosphere and will, after leaving the vessel, expand further, acquiring additional velocity in the process to a degree depending on its pressure of emergence. For the portion of the gas referred to above, I would suggest that the highest velocity present when atmospheric pressure is reached—the condition to which the energy equation refers—is of the order of 1,530 ft. per second, the first figure quoted by Dr. Geyer in his letter. During the subsequent part of the discharge the emergent velocity cannot exceed the now constant acoustic velocity of 892 ft. per second and will, in all probability, fall progressively below this figure as the discharge continues, thus giving an extremely wide range of velocities to be dealt with. May I suggest that the value of 1,060 ft. per second obtained by Dr. Geyer is something in the nature of a mean, or rather a root mean square, value for the whole of the escaped gases and that he is crediting the gas leaving the port at the instant of pressure equalisation with this velocity whereas it will really be moving very much more slowly?

Yours faithfully,
F. K. Bannister.

The University, Birmingham.
August 11, 1941.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. 56 MAY 1944 p.185 No. 2

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IMPROVING ENGINE PERFORMANCE BY EXHAUST PIPE TUNING.

By P. H. Schweitzer, Lt. Commander, U.S.N.R.

Perhaps the most neglected phase of two-stroke cycle engine design pertains to the exhaust systems. This neglect is fully reflected in the almost complete absence of American literature on the subject. This is the more deplorable as a properly designed exhaust system does not usually involve added weight or added complications and yet it improves engine performance all around. Even engines already installed could be improved by "tuning" their exhaust. Exhaust system design with regard to pressure waves is now in a status analogous to that of crankshaft design 20 years ago with regard to torsional vibrations. Considerable information has been discovered, but it has not spread widely enough and is generally ignored by the practical designers and installation men.

To be on the "safe" side, exhaust pipes are frequently oversized. But an oversize exhaust pipe is no better guarantee against undesirable synchronism than an oversize crankshaft is. Synchronism in torsional vibrations results in crankshaft breakage. Synchronism in pressure waves results in poor scavenging, reduced air charge, low power output, high fuel consumption and high exhaust temperatures. By avoiding the "criticals" the crankshaft is safe from torsional failure. By avoiding synchronism of pressure waves with engine speed we avoid poor filling of the cylinder and resultant power loss. But we can go one better by harnessing the pressure waves of appropriate frequency and thereby gain powerful assistance in scavenging and charging the cylinder. The result is higher output, lower fuel consumption, lower exhaust temperatures, lower piston and cylinder head temperatures, reduced maintenance and longer life. The difference between a well tuned and poorly tuned exhaust system is sometimes 30 per cent in engine output.

The exhaust system is frequently evaluated on the basis of the exhaust back pressure. But the exhaust back pressure as read on a manometer or pressure gauge attached to a certain point of the exhaust pipe is not significant because the actual pressure at any one point fluctuates during the cycle and the reading is only a rough average. Yet it is not this average which controls the charging and scavenging but the actual pressures during the scavenging period. The engine which has low (still better, sub-atmospheric) exhaust pressures during most of the scavenging period will give a better performance than one with high exhaust pressures, although the manometer reading of the exhaust pressure may be the same.

If no exhaust pipe were used, evidently the pressure at the exhaust opening would constantly be atmospheric. If a pipe is used, the sharp impulse initiated when the high pressure cylinder gases are put in communication with the exhaust pipe, sets up pressure waves of appreciable magnitude which travel back and forth in the pipe with the velocity of sound. The pressure next to the exhaust opening will rise and fall in accordance with the natural frequency of the system but with declining amplitudes until a new impulse from the subsequent opening of the exhaust is superimposed on the existing pressure waves. The effect of these pressure fluctuations on the scavenging and charging process may be favorable or unfavorable, depending on the timing of the waves, which, in turn, depends on the geometry of the exhaust system.

It is easy to see that if the natural frequency of the exhaust column is exactly equal to the number of engine revolutions per second, the pressure wave will so adjust itself that the rise will regularly coincide with the exciting impulse. This is undesirable, as it makes the pressure peaks in the exhaust duct coincide with the opening of the exhaust. The pressure wave will "buck" the exhaust and also the subsequent intake. The same will happen if the natural frequency of the exhaust is twice the revolutions per second.

In order to secure favorable exhaust conditions the period of natural oscillations should approximately equal the scavenging period. A partial vacuum will then exist during the latter part of the scavenging period, which is very helpful in drawing fresh air into the cylinder through the intake port. The depression should taper out when approaching exhaust closure, so that the scavenged cylinder may then fill up with fresh air rather than have this air sucked out again by the exhaust pipe vacuum.

Figure 1 shows three pairs of weak-spring indicator diagrams taken near the exhaust ports and in the crankcase of a crankcase-scavenged engine. The first pair shows unsatisfactory

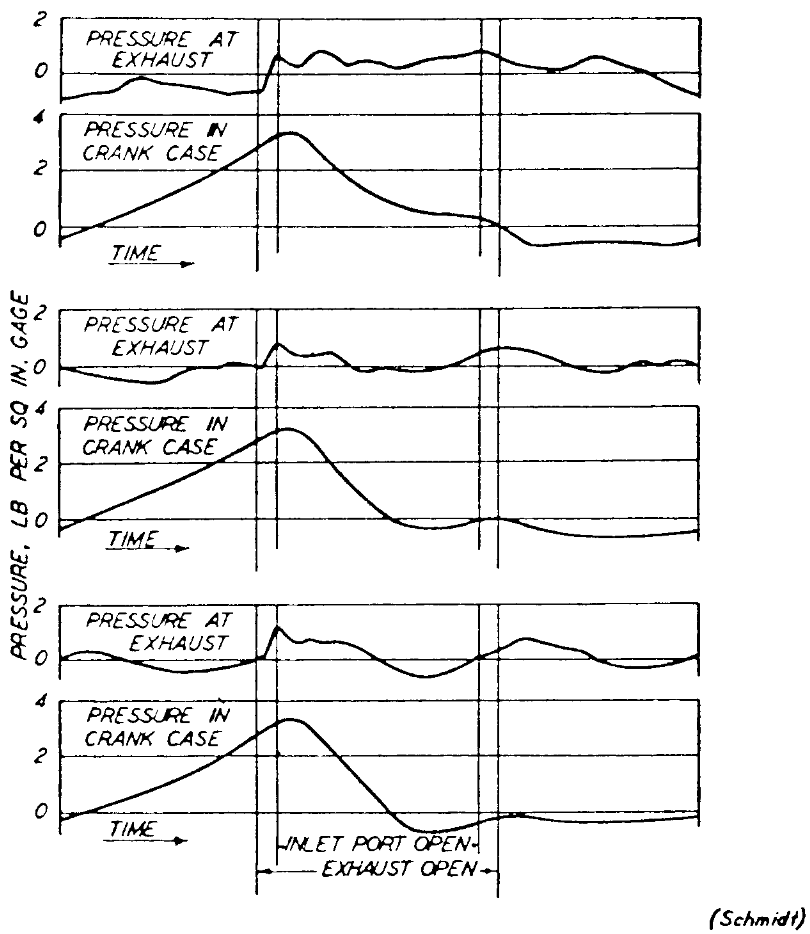

Figure 1.–Effect of Tuning on Pressure Fluctuations in the Exhaust Pipe at the Exhaust Port and in the Crankcase of a Crankcase Scavenged Engine During One Cycle. Top Diagrams Show Poor Bottom Diagrams Good, Middle Diagrams Indifferent Tuning.

scavenging. Since the pressure never drops below 0.4 pounds per square inch gauge during the whole scavenging period, the scavenged air cannot expand lower than to this value. Only after the closing of the ports does the pressure in the crankcase drop sufficiently to draw in fresh air from the outside. The pressure drop between crankcase and exhaust port is small or negative during the second half of the charging period which resulted in poor volumetric efficiency.

The third (bottom) pair of diagrams show very good scavenging. Note the vacuum at the exhaust near port closure. This causes a corresponding vacuum in the crankcase. The effect is that fresh air from the outside is drawn vigorously through the engine. The absolute exhaust pressure begins to rise when the piston is approaching port closure and depression. That helps to fill up the cylinder by the ramming effect of the air. The resulting high volumetric efficiency raised the thermal efficiency from 0.222 in the top diagrams to 0.252, an increase of 13.5 per cent. In the middle pair of diagrams conditions are in between. The only difference in the engine conditions represented by the diagrams was in the tuning of the exhaust system.

It may be in order to point out here that the "tuning" of the exhaust system depends primarily on its geometry, that means on the length and diameter of the exhaust pipes and on the volume of the various containers attached or interposed in the exhaust system. If an exhaust system is tuned it stays tuned irrespective of load and operating conditions. The only exception is a change in speed. An engine can be tuned for one speed only, and, therefore, tuning has the greatest significance for constant speed engines. Variable speed engines should be tuned to the speed at which optimum performance is desired. This may be the most common operating speed or the speed of maximum power output.

Being aware of the importance of the exhaust tuning, the practical problem is how to create favorable exhaust conditions for a given engine at a given speed. Three methods will be described in this article: 1. Performance test method; 2. Analytical method; 3. Pressure indicating method. Often a combination of the methods will be found advantageous.

Performance Test Method.

Exhaust tuning affects such important performance characteristics as maximum power output, specific fuel economy, air

delivery and exhaust temperature. In a most direct method, therefore, we would measure some such performance characteristics while the tuning of the exhaust system is being varied. Then we select the tuning which gives best engine performance.

This method can be applied very successfully to a small engine. Belilove (Ref. 1) measured the air delivery ratio (volume of air delivered to the cylinder divided by the piston displacement) and the power output of a 3/4 Hp. Evinrude outboard engine while he varied the length of the 3/4 inch exhaust pipe (inside diameter 13/16 inch), and obtained Figure 2 (6). It was simple to conclude that a 28 inch long pipe was the best for that engine at 2800 Rpm.

For large engines it is not always feasible to measure power, air consumption or specific fuel consumption because of lack

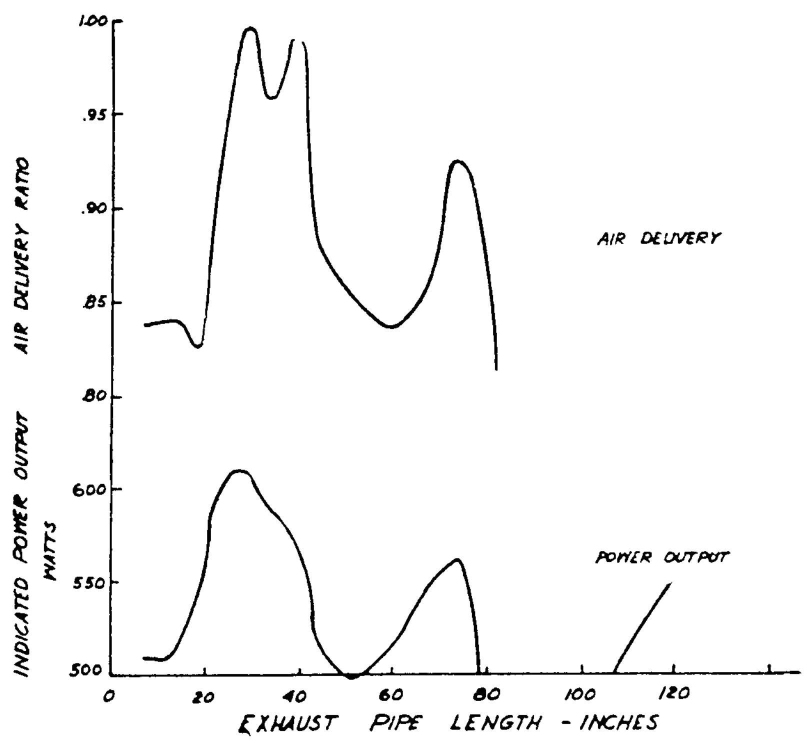

FIGURE 2.—POWER AND AIR DELIVERY WITH AN EVINRUDE $1\frac{3}{8}$-INCH × $1\frac{3}{8}$-INCH SPARK IGNITION ENGINE AT 2800 RPM.

of equipment. The measuring of the exhaust temperature alone without power measurement is inadequate. Making numerous changes on a bulky exhaust system is costly and time-consuming

Another unattractive feature of this method is that one or two measurements do not disclose how near the system is to optimum tuning and in which direction the optimum will be found. Therefore, the principal role of this method lies in checking the results obtained by other means.

ANALYTICAL METHOD.

Taking advantage of such theoretical work as was done by Schmidt (Ref. 2) and Zeman (Ref. 3) in Germany, and Farmer (Ref. 4) and Mucklow (Ref. 5) in England, the problem of exhaust tuning can be made accessible to analytical calculations. In this manner the exhaust system can be tuned before it is built, saving much time and expense. In the following, methods of calculating exhaust systems are to be presented in a simplified manner, ignoring the derivations of the formulae and preceding theoretical work.

The frequency of the gas column oscillations in the exhaust system is determined primarily by the length of the exhaust pipe and secondarily by its diameter and the volumes interposed in the system. Therefore, it is reasonable to start the calculations by selecting the diameter of the main exhaust pipe. If the exhaust pipe cross section is inadequate the exhaust will be throttled irrespective of its tuning. If it is too large, the amplitude of the pressure waves will be small and the effect of tuning will thereby be minimized. That is an advantage if the tuning is incorrect but a disadvantage if it is correct. The tendency in the past has been to use oversized exhaust pipes for two-stroke cycle engines, presumably just to be on the safe side and avoid the effects of the "incalculable" pressure waves.

There is little beyond thumb rules to guide us in the selection of the diameter of the exhaust pipe. To avoid throttling, the gas velocity in the pipe or duct must be lower than in the exhaust ports, preferably a third less. In multi-cylinder engines the gas velocity in the common header or exhaust pipe should be still lower. Burgess Battery Co., makers of exhaust snubbers, recommend 50 feet per second for crankcase scavenged engines; from 65 to 115 feet per second for low-speed (up to 350 Rpm.) separately scavenged engines; from 100 to 150 feet per second for medium-speed (350 to 1200 Rpm.); and from 135 to 165 feet per second for high-speed (above 1200 Rpm.) two-stroke cycle engines. In calculating the conduit size from the permissible gas velocities, it must be taken into account that the volume of the exhaust gas is about double the volume of the

intake air, based on the ratio of absolute temperatures.

Example: A 16-cylinder 8-1/2x10-inch 800 Rpm. engine has an air-delivery ratio of 1.325 and a gas temperature (in the exhaust pipe) of 500 degrees F. at full load. What is the required size for the exhaust pipe? The exhaust gas volume is

$$1.325\times16\times\frac{8.5^2\pi}{4}\times10\times\frac{800}{1728}\times\frac{460+500}{460+150}=8700$$

cubic feet per minute.

Allowing an average gas velocity of 150 feet per second, the cross section of the common exhaust pipe must be

$$A=\frac{8700\times1728}{60\times150\times12}=139 \text{ square inches}$$

so that the inside diameter of the conduit is to be 13.3 inches. The next standard pipe size is 14 inches nominal inside diameter.

After the size of the exhaust pipe has been selected, its tuning will be effected by adjusting its length and the volumes interposed in the system. The controlling factor in the calculation is the natural frequency of the pressure waves.

Pressure Waves.

The pressure waves in the exhaust pipes are similar to the sound waves in organ pipes and are controlled by identical laws. In a plain pipe closed at both ends the period of the pressure waves is 2 L/a where L is the length of the pipe and a is the velocity of the sound in the gas. The value of a varies with the gas temperature according to the formula

$$a=C\sqrt{k\,p\,v}=C\sqrt{k\,R\,T}$$

where k is the ratio of the specific heats of the gas, p, v, and T its mean pressure, specific volume and absolute temperature in the pipe, and C a constant which decreases in straight line relationship inversely as the bore of the pipe. According to the most recent determinations (Ref. 1) Figure 3 represents the velocity of pressure propagation in pipes with average composition of exhaust gas for various mean exhaust gas temperatures and pipe diameters. The statement on page 189 that the tuning is independent of the load needs correction in so far that a change in load changes the exhaust temperature which changes to some extent the propagation velocity of the pressure wave.

For an exhaust system consisting of a single pipe of uniform cross section attached at one end of the exhaust ports, the other end being open to the atmosphere, the period of gas column vibration is about double, that means 4 L/a. The reason for this is that the pressure wave is reflected at the open end of the pipe with sign reversed. The period of this negative wave is also 2 L/a and, therefore, the total time of the complete cycle is 4 L/a.

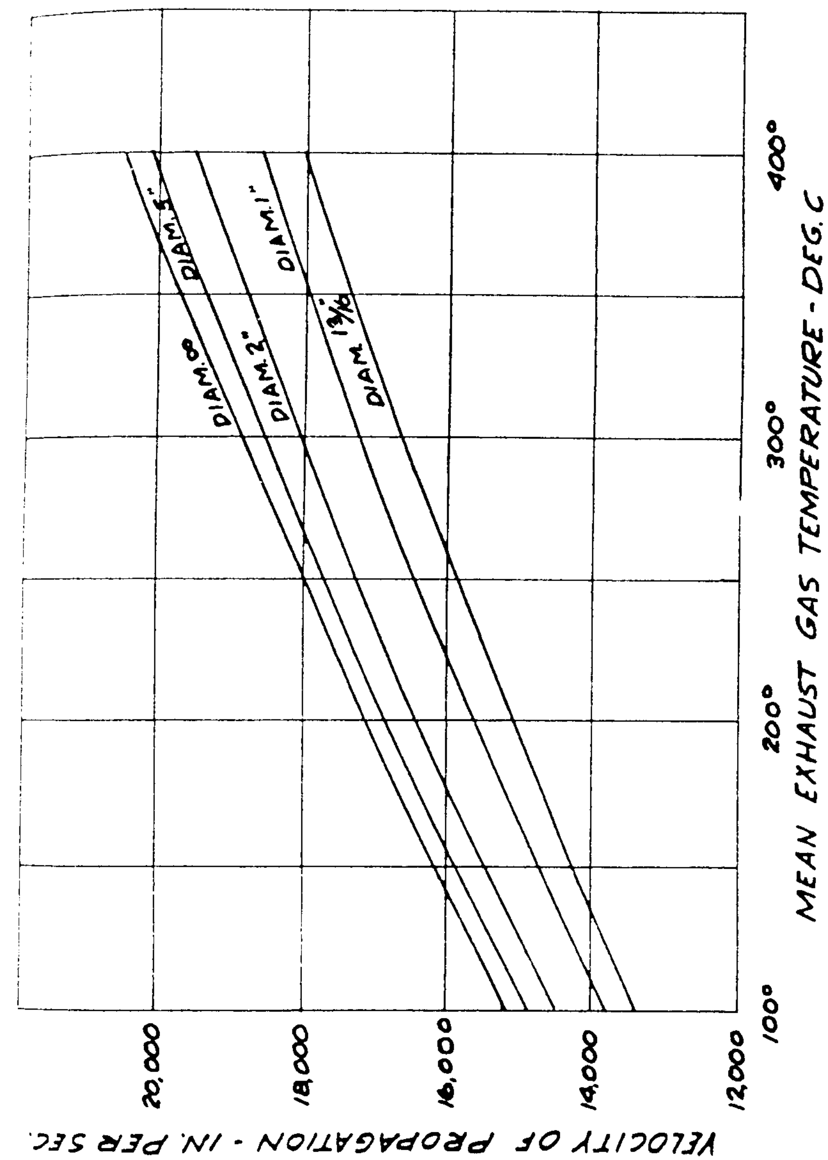

Figure 3.—Variation of Propagation Velocity of Pressure Waves in Pipes of Various Diameters.

In calculation of pressure waves in pipes it is customary in replace the oscillating system with a pipe of length L_e of uniform cross section closed at both ends which has the same frequency or period as the oscillating system. This is called the equivalent pipe length. For a plain exhaust pipe of uniform cross section throughout and open to the atmosphere, the equivalent pipe length is

$$L_e = 2(L + C_R)$$

where L is the actual length of the pipe and C_R the so-called Raleigh correction. Since the reflection does not take place exactly at the open end, an additional length roughly equal to 0.4 times the inside diameter of the pipe is added to the actual pipe length. Except with very short pipes the Raleigh correction is relatively small and may be neglected, therefore

(1) $$L_e = 2L$$

Using the convenient concept of the equivalent pipe length the complete natural period of the vibration of the gas column always is

(2) $$t = \frac{2L_e}{a}$$

Consequently, if the exhaust port of the engine is connected through a plain pipe of the length L to the atmosphere, the period of the exhaust column vibration will be $t = 4\ L/a$.

Example: The single cylinder 1-3/8 inches x 1-3/8 inches Evinrude engine mentioned above has (see Ref. 1) its exhaust ports open at 107 degrees after top center and closed at 253 degrees top center, therefore a port opening period of 146 degrees. The 13/16 inch i.d. exhaust pipe is directly attached to the exhaust ports. What will be the worst length for the exhaust pipe and what will be the best for 2800 Rpm?

Assuming a mean temperature of 175 degrees F. in the exhaust pipe, Figure 3 gives a propagation velocity of 14,700 inches per second.

The worst gas column frequency will be the one which is equal to the engine frequency, whose period is

$$t_w = \frac{60}{2800} = \frac{10}{465} \text{ second}$$

The equivalent pipe length which gives this frequency to the system is from Equation (2)

$$L_{ew} = \frac{a\,t_w}{2} = \frac{147000}{2 \times 465} = 158.5 \text{ inches}$$

The actual pipe length will be from Equation (1)

$$L_w = \frac{L_{ew}}{2} = \frac{158.5}{2} = 79.25 \text{ inches}$$

Referring to Figure 2 we note that the power and air delivery curves take at 80 inch pipe length a steep dive.

The best gas column frequency should be the one which gives a period equal to the port opening period. The latter is

$$t_b = \frac{146}{360}\ \frac{60}{2800} = \frac{1}{111} \text{ second}$$

The equivalent pipe length is

$$L_{eb} = \frac{a\,t_b}{2} = \frac{13800}{2 \times 111} = 62 \text{ inches}$$

(The lower propagation velocity of 13,800 inches per second corresponds to the lower mean exhaust gas temperature of 125 degrees F. which is more likely with optimum tuning.) The best actual pipe length then is

$$L_b = \frac{62}{2} = 31 \text{ inches}$$

Inspecting Figure 2, we find this value to check with the maximum power and air delivery.

Frequently the location of the engine is such that the desirable pipe length is not sufficient to reach the outside atmosphere. In such case a large exhaust pit or expansion chamber can be used in place of the atmosphere at the end of the exhaust pipe, and the expansion chamber can be connected to the atmosphere by a tail pipe. The length of the latter is immaterial and will have no effect on the pressure waves in the primary exhaust pipe, provided that its cross section is large enough to keep the pressure in the expansion chamber substantially atmospheric. A rule will be given later in the text for the size of the expansion chamber and the minimum diameter of the tail pipe.

However, ordinarily the exhaust pipe does not connect to the exhaust ports directly, but a duct, sleeve or chamber of a certain volume is placed between the exhaust ports and the exhaust pipe. The arrangement is shown in Figure 4 (a) when the exhaust connects directly into the open and Figure 4 (b) when it connects into an expansion chamber. The volume V_1 next to the exhaust ports will affect the exhaust tuning in either

case and we will refer to it as the "exhaust pot" even if it consists only of a small enlargement over the exhaust pipe cross section. The size of the expansion chamber V_2 and tail pipe does not affect the tuning if both are large enough to keep the pressure in the expansion chamber substantially atmospheric.

The equivalent pipe length of such a system has been calculated by Thomas Schmidt (Ref. 2) and can be expressed by the following formula:

$$\tan\frac{\pi L}{L_e} = \frac{A L_e}{\pi V_1} \qquad (3)$$

where L is the actual, L_e the equivalent pipe length, A is the cross sectional area of the exhaust pipe and V_1 the volume of the exhaust pot close to the engine. Chart, Figure 5, shows the relation in graphical form.

It will be noted that equivalent pipe length (and therefore, the natural frequency of the exhaust system) is determined not by the pipe length alone, but also by its cross section and the volume between the engine and the exhaust pipe. Even a small volume V_1 increases the equivalent pipe length considerably. For instance, in the example treated above an exhaust pot of only 26 cubic inches between the engine and a 25 inch exhaust pipe will increase the equivalent pipe length from 51 inches to 121 inches.

In using Formula (3) or Chart 5, the exhaust pot volume V_1 should include any enlargement found beyond the exhaust port such as ducts, sleeves, etc., but only the volume in excess of the corresponding exhaust pipe should be counted. The actual exhaust pipe length should be counted from the cylinder to the atmosphere or the large expansion chamber, and to that the Raleigh correction of 0.4×I.D. may be added.

Example: A 7.9×11.8-inch, one cylinder, 18-Hp. crankcase-scavenged engine operating normally at 370 Rpm. has an exhaust opening period of 136 degrees crank angle. The exhaust ports connect directly into an exhaust pot of 2000 cubic inch volume. From this an exhaust pipe of 5.35 inches I.D. leads to the atmosphere. What is the worst exhaust-pipe length, and what is the best?

The worst frequency is that equal to the engine frequency which corresponds to a period of

$$t_w = \frac{60}{370} = \frac{1}{6.15} \text{ second}$$

The corresponding equivalent pipe length is from Formula (2)

$$L_e = \frac{a\,t_w}{2}$$

With an estimated mean exhaust gas temperature of 210 F., the propagation velocity of the pressure waves from Figure 3 is 14,900 inches per second and, therefore,

$$L_e = \frac{14900}{2 \times 6.15} = 1210 \text{ inches}$$

The ratio $V_1/A = 2000/22.5 = 89$ inches, which gives through Chart 5 an actual pipe length of 515 inches less $0.4 \times 5.35 = 2.14$ inches Raleigh correction, therefore

$$L_w = 513 \text{ inches}$$

This would be the worst pipe length.

The best frequency is that which corresponds to the period of exhaust duration, which is

$$t_b = \frac{136}{6 \times 370} = \frac{1}{16.33} \text{ seconds}$$

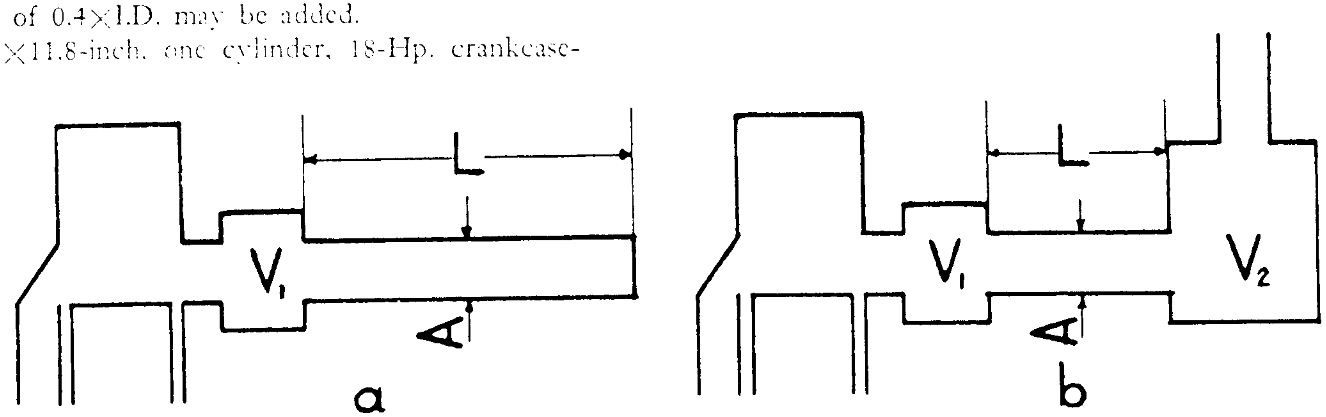

Figure 4.—Arrangement of Exhaust Systems.

and the corresponding equivalent pipe length

$$L_e = \frac{a\, t_0}{2} = \frac{14900}{2 \times 16.33} = 456 \text{ inches}$$

Chart 5 gives now with the same $V_1/A = 89$ inches, 150 inches and applying again the Raleigh correction

$$L_e = 148 \text{ inches}$$

as the best pipe length.

It so happens that Schmidt has tried out a number of pipe lengths with the above engine and accurate records were taken. He found a 490 inch pipe length very bad and a 132 inch length very good. This shows that the calculation is not perfect, probably because of the uncertainty of estimating the mean exhaust temperature of the line, the existence of residual pressure waves and other minor factors. Nevertheless, with a simple exhaust system the calculation can be depended on to give fairly close results.

Large Expansion Chamber.

The above calculation is correct for an exhaust system such as shown schematically on Figure 4 where the main exhaust pipe connects to the atmosphere or to an expansion chamber sufficiently large to maintain substantially atmospheric pressure at the end of the exhaust pipe. This expansion chamber ordinarily connects through a tail pipe to the atmosphere and this tail pipe must be, of course, of such size as to prevent throttling of the exhaust gases. The following rules should help to select the size of the tail pipe and of the expansion chamber.

The manometric back pressure in the expansion chamber should not exceed 10 inches water. The pressure drop per 100 feet of pipe can be expressed as

$$h = \frac{Q^2 L}{4430\, d^5} \text{ inches water}$$

where Q is the volume of the exhaust gas cubic feet per minute, d the diameter of the tail pipe in inches and L its length in feet.

For the size of the expansion chamber we set the arbitrary requirement that it should not change the natural period of the exhaust system by more than 5 per cent. Conforming to this requirement Figure 6 shows the minimum ratios V_2/V_1, for various pipe lengths and V_1/F ratios. Accordingly the secondary

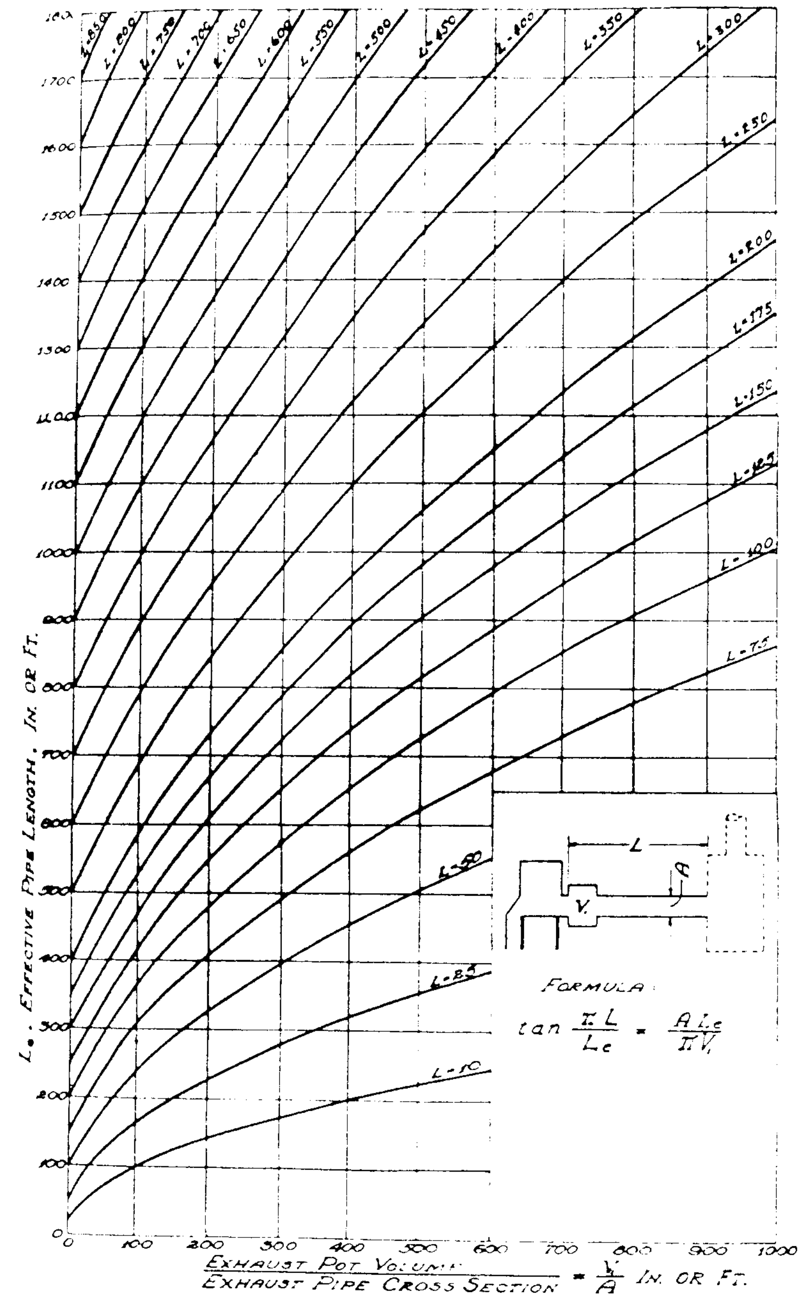

Figure 5.—Charts Showing the Equivalent Pipe Lengths for a Frequently Used Exhaust System.

exhaust pot or expansion chamber must be at least about 10 times larger than the primary exhaust pot in order to be equivalent to the atmosphere and permit the use of formula and chart shown in Figure 5.

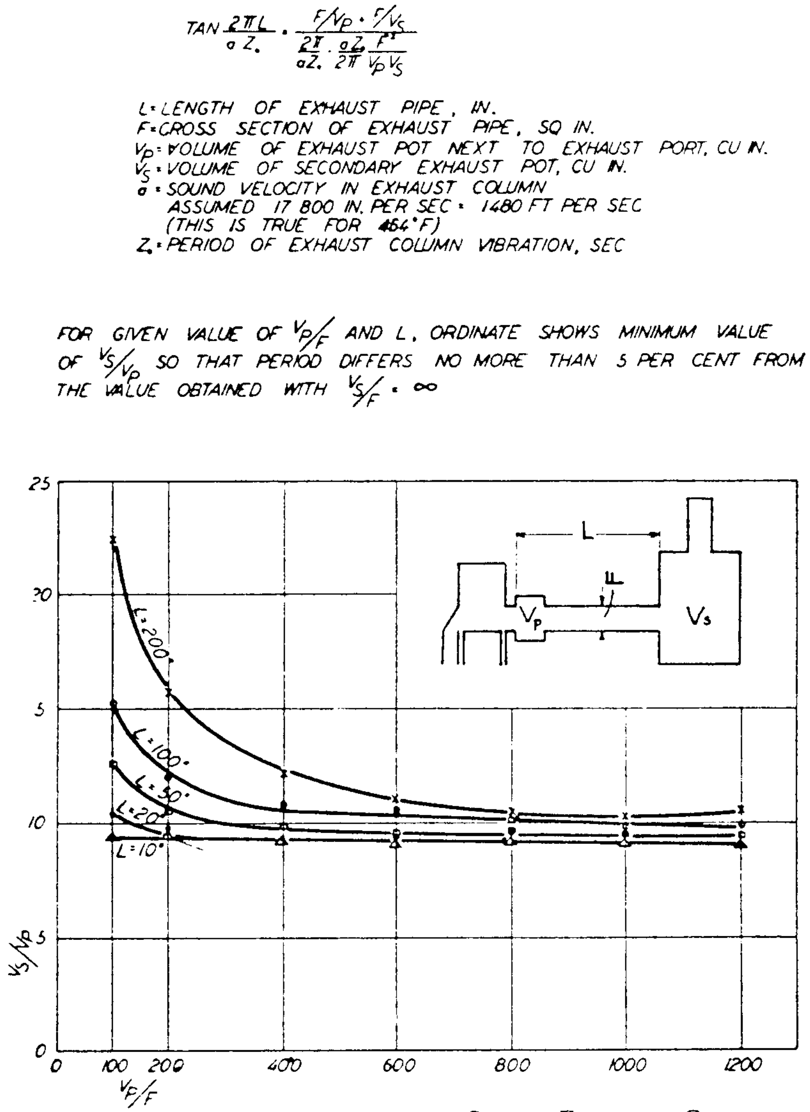

Figure 6.—Chart Showing Selection of Size of Expansion Chamber.

Small Expansion Chamber.

When secondary exhaust pot or expansion chamber is relatively small (see Figure 7), it will affect the natural frequency of the oscillating system. Schmidt (Ref. 2) developed the calculation for this case also and the resulting formula is as follows:

$$\tan\frac{\pi L}{L_e} = \frac{\frac{A}{V_1} + \frac{A}{V_2}}{\frac{\pi}{L_e} - \frac{L_e}{\pi}\,\frac{A^2}{V_1 V_2}}$$

This formula is rather complicated and its use is preferably avoided by increasing the relative size of V_2.

Another system is shown in Figure 8 with a single exhaust pot in the middle of the exhaust pipe, for which the following equation is valid:

$$\operatorname{ctn}\frac{\pi L_2}{L_e} - \tan\frac{\pi L_1}{L_e} = \frac{\pi V}{A\,L_e}$$

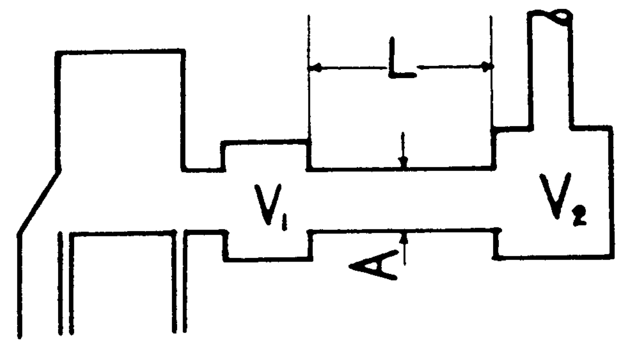

Figure 7.—Exhaust System Including a Relatively Small Expansion Chamber.

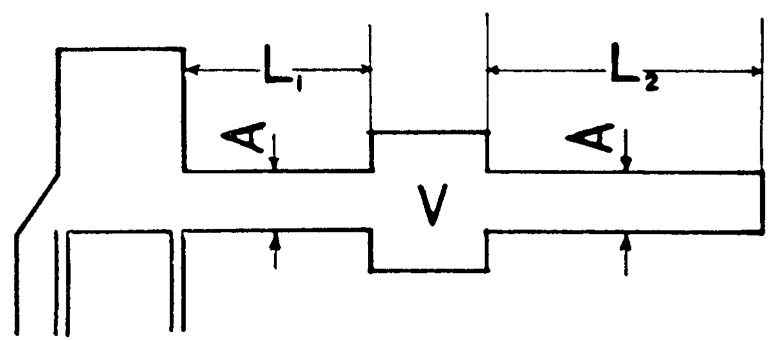

Figure 8.—Exhaust System With an Exhaust Pot in the Middle of the Exhaust Pipe.

Example: In Figure 9, four pressure diagrams of the exhaust pipe are shown, presenting successively deteriorating tuning. The only variable in the setup was the size of the expansion chamber V_2. In order to equal the performance with no expansion chamber, Schmidt had to supply an expansion chamber of 0.6 m^3=36,000 cubic inches. According to Chart 6 the expansion chamber would have to be 17 times the primary exhaust pot or 17×2000 cubic inches=34,000 cubic inches, which checks closely with the experimental value. By reducing the secondary

volume to 13,300, 8600 and 1840 cubic inches respectively, diagrams b, c and d were obtained and performance deteriorated, as it is shown by the indicated figures on specific fuel consumption and exhaust temperature. The power output was kept substantially constant during the series.

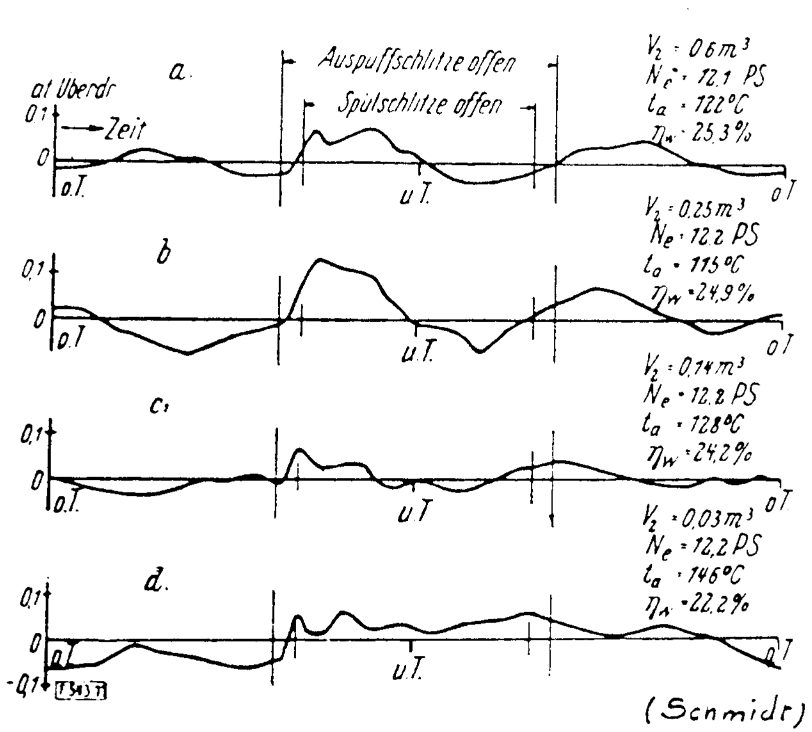

FIGURE 9.—INDICATOR DIAGRAMS SHOWING THE EFFECT OF SIZE OF EXPANSION CHAMBER.

If the optimum exhaust pipe length is too short for structural reasons, even when used in conjunction with an expansion chamber, favorable results can also be obtained by multiplying the equivalent pipe length by two.

Example: A 12×15 inch, 367 Rpm. crankcase scavenged engine had an 8 inch I.D. exhaust pipe discharging into the open. It had a 78-1/2 inch long muffler, close to the engine, with a volume of 8550 cubic inches. The average exhaust gas temperature was estimated at 464 degrees F. and the exhaust per opening period 135 degrees crank angle.

The exhaust-pipe cross section was $8^2\pi/4=50.3$ square inches. Since the muffler was in the line, the replaced pipe volume $50.3\times78.5=3950$ cubic inches had to be deducted, giving an effective exhaust pot volume of $V_1=8550-3950=4600$ cubic inches and $V_1/A=4600/50.3=91.5$ inches.

The worst frequency is the one equal to the engine frequency, which had a period of $t_w=60/367=1/6.125$ second. With a sound velocity of 17,800 inches per second, the equivalent pipe length is

$$L_e=\frac{a\,t_w}{2}=\frac{17,800}{2\times6.125}=1452 \text{ inches}$$

From Formula 3 the actual pipe length corresponding to this with $V_1/A=91.5$ inches is

$$L_w=640 \text{ inches}$$

This would be the worst pipe length. For best pipe length the required period is

$$t_b=\frac{135}{6\times367}=\frac{1}{16.25}\text{second}$$

which would give an equivalent pipe length of

$$L_e=\frac{a\,t_b}{2}=\frac{17,800}{2\times16.25}=550 \text{ inches}$$

or an actual pipe length of 190 inches, which was too short to reach from the building to the open; therefore, the equivalent pipe length is doubled, making 1100 inches. With this the actual pipe length turns out to be 460 inches as most favorable.

According to tests made by K. C. Whitefield (Ref. 6) on this engine, the worst pipe length was 600 inches and the best 456 inches, which is very good agreement, indeed.

MULTICYLINDER ENGINES.

The above formulae are strictly applicable only for either single cylinder engines or multicylinder engines with individual exhaust pipes to the atmosphere or to a large expansion chamber. The last arrangement is quite common with an exhaust pit under the floor into which the individual exhaust pipes connect. If the volume of the exhaust pit exceeds the size determined by Figure 6, and it connects with a large enough tail pipe to the open, the multicylinder engine can be treated as so many single cylinder engines as far as exhaust tuning is concerned.

Frequently, however, the individual exhaust pipes combine by tee's or Y's before reaching the atmosphere or expansion chamber. In other cases no individual exhaust pipes are used at all, only a common header or exhaust manifold. These cases

are not easily accessible to calculation because the exhaust impulse from one cylinder interferes with the residual pressure waves set up by another cylinder. To evade this difficulty the exhaust manifolds sometimes combine not more than two or three cylinders the firings of which are at least 120 degrees crank angle apart and these manifolds in turn are led into an exhaust pit or expansion chamber. The complication of such an arrangement with a large number of cylinders is readily understandable.

When exhaust headers or manifolds are used in a multi-cylinder engine, it is advisable not to depend on calculations alone for tuning. The indicating method on the other hand is applicable to any number of cylinders and its intelligent use should pay handsome dividends.

Indicating Method.

It was pointed out that the pressure record near the exhaust port of a well tuned exhaust looks like Figure 9 *a*, where we see a nice negative loop ending at about the exhaust closure. Less good tuning is shown in diagram *b*, where the negative loop is still pronounced but ends a little too early. In diagram *c* the negative loop almost vanished during the scavenging period, but the worst tuning is shown by diagram *d*, where positive pressure persists during the whole scavenging period.

Therefore, if reliable pressure diagrams of the exhaust of each cylinder are available, by their inspection the quality of the exhaust tuning can easily be determined. If the tuning is not satisfactory, slight changes in the geometry of the exhaust system (size and lengths of ducts and pipes, volumes of ducts, chambers and silencers) should improve it. The shape of the pressure diagram indicates even the direction in which the improvement is to be sought. If the negative loop is too short, the period of oscillation must be increased by adding length and/or volume. If the negative loop is too long, the period must be shortened by decreasing length and/or volume. If for each cylinder the negative loop ends at about exhaust closure, we have perfect tuning.

Taking of exhaust pressure indicator diagrams is not easy with conventional indicators. Mechanical weak-spring indicators are suitable only for relatively low speeds. The diagrams shown in Figure 10 were taken by Belilove (Ref. 1) with a cathode ray oscillograph. An electromagnetic pickup specially fitted with a thin steel diaphragm of 0.0035 inch thickness was attached to the exhaust pipe near the exhaust ports and connected to a 9 inch RCA cathode ray oscillograph to which an integrator and a degree marker was added. The wiring circuit of the integrator and degree marker is shown in Ref. 7.

The records obtained with the Evinrude engine mentioned above check satisfactorily with the performance tests.

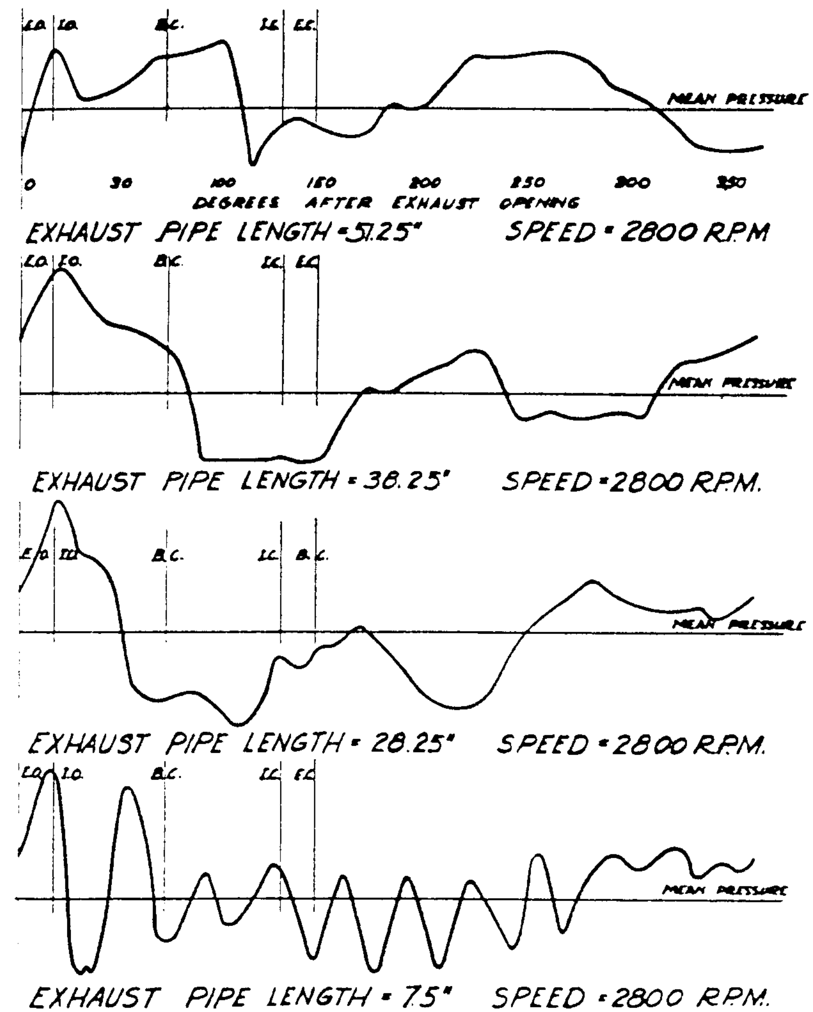

Figure 10.—Pressure Time Diagrams Obtained by a Cathode Ray Indicator. (The Carbon Pile Pressure Element Was Placed in the Exhaust Pipe Next to the Port.)

Nevertheless it was found that a cathode ray indicator is not very suitable for low pressure indication, especially because of the uncertainty of the pressure scale and the zero line.

On the other hand the instrument shown in Figure 11 has proven itself very satisfactory for indicating exhaust pressures.

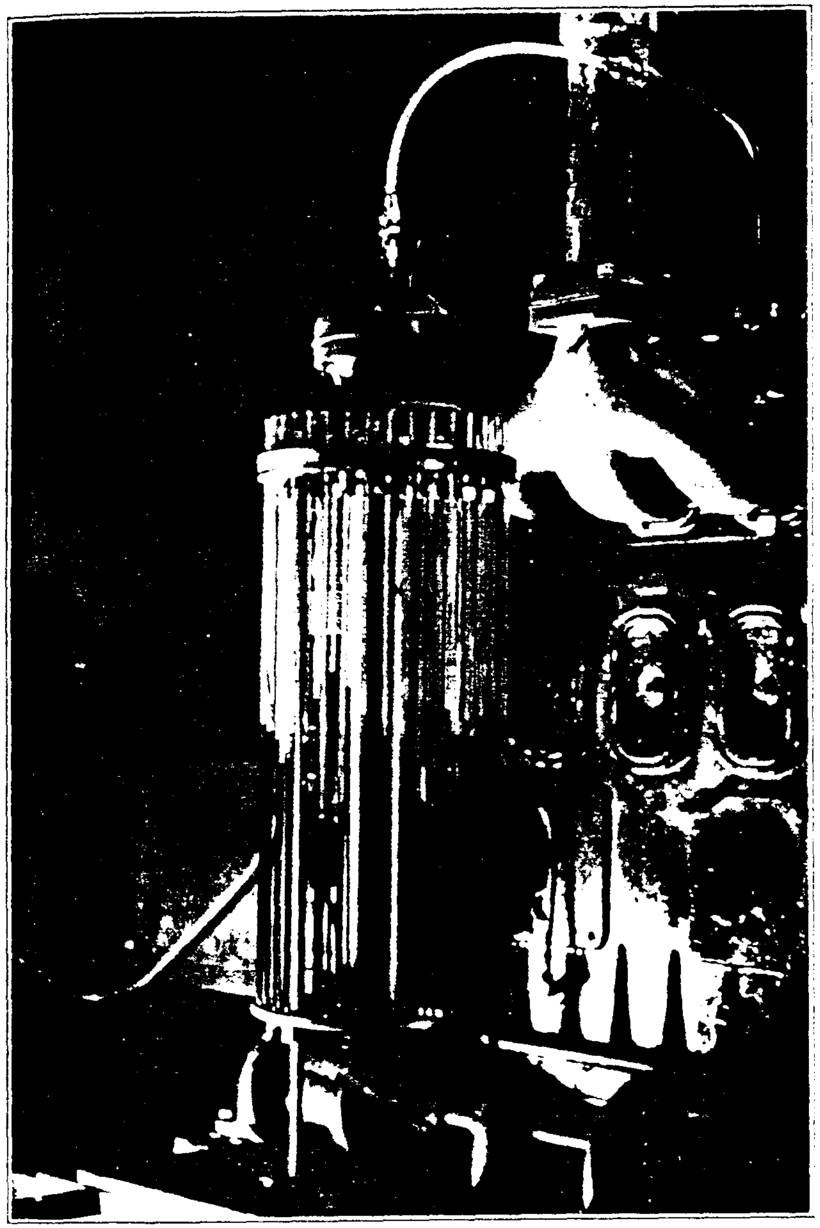

FIGURE 11.—PENN STATE ROTARY VALVE LOW PRESSURE INDICATOR.

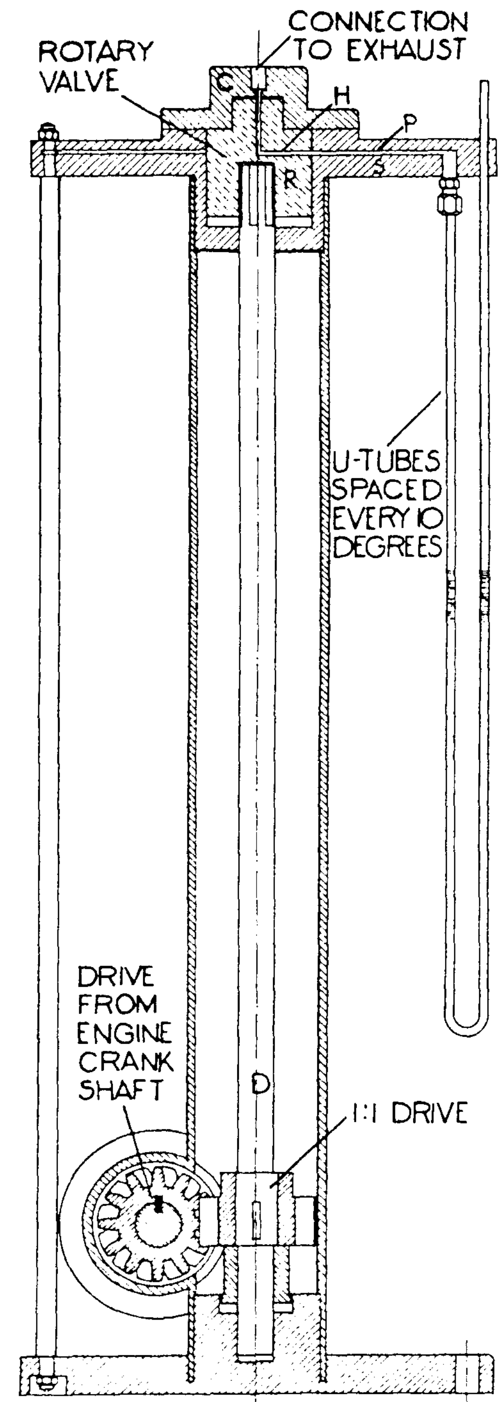

FIGURE 12.—SCHEMATIC DRAWING OF PENN STATE ROTARY VALVE LOW PRESSURE INDICATOR.

It has been developed recently in the Diesel Laboratory of The Pennsylvania State College.

Figure 12 is a schematic drawing of the indicator. It consists of a rotor R, with a single distributor hole H, driven from the engine crankshaft through spline driveshaft D. The rotor rotates inside of stator S which has 36 equally spaced passages P, and brings every ten degrees another manometer tube, M, in connection with the exhaust port. The connection to the exhaust (or other space the pressure of which is to be indicated) is through a tubing attached to the stationary cover plate C, above rotor R, the distributor hole of which remains in continuous connection with the connecting tubing. The connection of the indicator to the exhaust port of a 3-71 General Motors engine is shown on Figure 11. To avoid soot from the exhaust getting in the manometer tubes, a thin rubber membrane is interposed between the exhaust and the cover plate of the rotary valve as visible on Figure 11. The manometer tubes corresponding to 0, 10, 20, etc. degrees of crank rotation are identified by a rotatable band type degree scale (Figure 11) near the top of the manometer tubes, which can be adjusted to proper phase relation with the crankshaft.

The rotor is lap fitted in the stator and cover plate which insures a good seal and makes leakage loss negligible. Since mercury is too heavy for accurate reading of exhaust pressures, while water is too light, bromoform (specific gravity 2.87) has been used, with a drop of methyl orange added for better visibility.

Figure 13 is a sample record obtained with the instrument which shows fairly good exhaust conditions for No. 1 cylinder, less good for No. 2 and poorest exhaust for No. 3 cylinder. The exhaust temperature reading was highest for No. 3 and lowest for No. 1 cylinder.

In examining the exhaust pressure records of multicylinder engines, it must be kept in mind that with a common exhaust header or manifold it is seldom possible to create perfect tuning for each cylinder, as the exhaust impulses necessarily intermingle and the exhaust of one cylinder even "backfires" into the open exhaust of another cylinder which succeeds the former in firing order. Various schemes are being used to minimize the effect of undesirable pressure fluctuations in the common header of multicylinder engines, by properly placed mufflers and silencers, tapered exhaust nipples and other arrangements calculated to

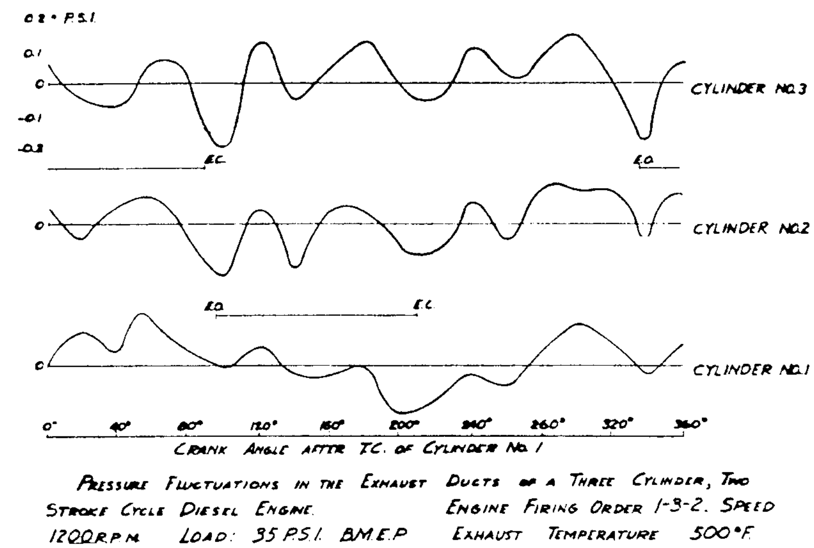

FIGURE 13.—EXHAUST PRESSURES OF A G.M. 3-71 TWO-STROKE CYCLE ENGINE FOR EACH OF THE THREE CYLINDERS.

dissipate the pressure energy of the exhaust, and prevent it from interfering with the scavenging process. The exhaust conditions in a multicylinder engine with a common exhaust header are, therefore, never ideal. We must be satisfied if the exhaust pressure indicator shows the absence of high instantaneous pressures in the exhaust ducts during the respective scavenging period.

In order to effectively tune a multicylinder engine, individual exhaust pipes of tuned lengths must be mounted between the individual cylinders and the common header or exhaust pot. As an alternative, two or three cylinders may exhaust into a single exhaust pipe, provided the firing order is such that the exhaust periods of the cylinders exhausting into a common pipe do not overlap.

In conclusion, it may be in order to point out that marine applications offer a highly fruitful field for applying exhaust tuning. Auxiliary units for generating electrical power, whether AC or DC, are run at constant speed. Some main propulsion engines have electrical drives. For engines mechanically connected to the propeller, the power requirement varies approximately as the cube of the rotating speed and sharp tuning for maximum speed is desirable, because that is the only occasion when maximum power is needed.

Any two-stroke cycle engine, single or multicylinder, with a properly tuned exhaust system, delivers more power than the same engine would if exhausting into the open with zero length

exhaust pipes. Unfortunately, most engines in service deliver considerably less power than they would with zero exhaust pipes. Neutralizing the interference of improperly tuned pressure waves increases the power output and reduces the exhaust temperature of some cylinders and of the entire engine. This can be accomplished by suitable exhaust ducts, large exhaust headers, and various damping devices. Best results, however, are obtained by a correctly tuned exhaust system when the pressure waves actively assist in scavenging and cylinder charging.

Conclusions.

1. By proper tuning of the exhaust the performance of two-stroke cycle engines can be appreciably improved.

2. The use of large exhaust pipe diameters and high scavenge pressures are often not justified except on the basis that they tend to minimize the effects of the "incalculable" pressure waves.

3. For many installations the natural frequency of the exhaust system can be calculated and the optimum pipe lengths predicted from design data.

4. Multicylinder engines with common exhaust headers or manifolds also can be improved by proper exhaust tuning but no mathematics is available to handle the problem.

5. With the use of an exhaust pressure indicator, the tuning of any engine may be improved. A recently developed rotary valve indicator should greatly facilitate experimental exhaust tuning.

6. For best tuning, a multicylinder engine should have individual exhaust pipes for each cylinder, or for each two or three cylinders whose exhaust periods do not overlap.

7. If correct tuning of the exhaust system is out of reach, the neutralizing of the interference of improperly timed pressure waves usually improves engine performance.

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2. T. Schmidt. Schwingungen in Auspuffleitungen von Verbrennungsmotoren. V.D.I. Dieselmaschinen. VI, 1936, p. 79.

3. J. Zeman. Baugrenzen von Zweitakt-Dieselmaschinen mit Kurbelkasten Spuelpumpe, V.D.I. Dieselmaschinen VI, 1936, p. 142.

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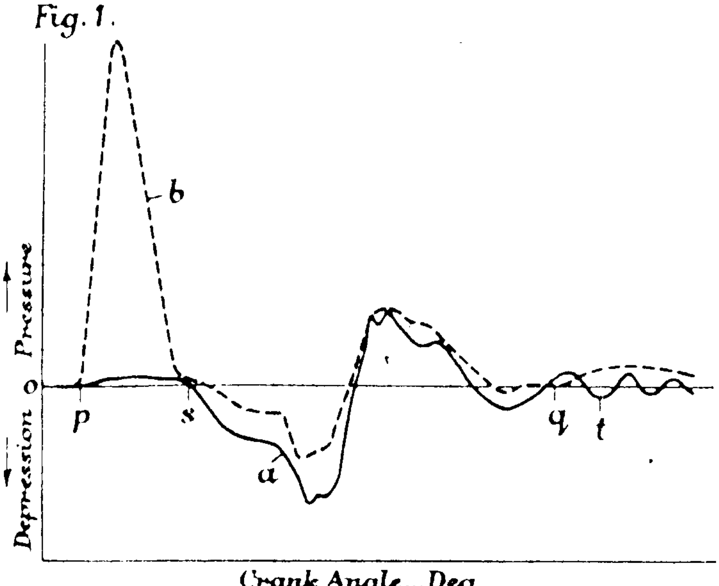
7. P. H. Schweitzer. Use of the Oscillograph for Testing Fuel Injection. Automotive Industries. January 1, 1942.

THE KADENACY EFFECT.

By P. H. SCHWEITZER, C. W. VAN OVERBEKE and L. MANSON.

THE Kadenacy system of scavenging is best known in application to a blowerless two-stroke engine. The inertia of the exhaust gases is utilised to produce a depression or partial vacuum in the cylinder, and the fresh charge is introduced by the suction thus created. The Kadenacy system; however, apparently embraces more than this. In his numerous patents, Kadenacy claims that the burnt gases discharge from the cylinder substantially "as a mass in an interval of time shorter than that required for the burnt gases to expand down to the ambient atmospheric pressure by adiabatic flow," and the depression so created is utilised for introducing the fresh charge through an inlet which is opened while the issuance of the burnt gases is in full progress. Tests reported by P. W. Petter,* by S. J. Davies,† and others‡ show astonishing results and seem to indicate that a new principle is involved which is responsible for the unusual effect. The most remarkable aspect of the published results is that the effect persists over a wide speed range—800 r.p.m. to 1,700 r.p.m. in case of a converted Junkers opposed-piston engine—and over greatly varying exhaust-pipe lengths.

Fig. 1.



* *The Engineer*, vol. 158, page 157 (1934).
† ENGINEERING, vol. 143, page 685 (1937).
‡ *Gas and Oil Power*, November, 1939.

The mechanics of the Kadenacy principle were investigated by S. J. Davies with a free-moving piston in a cylinder. He compressed a mixture of air and petrol vapour with a hand crank and ignited it with a sparking plug. The gas pressure drove the piston downward until it uncovered a slot through which the burnt gases discharged into the atmosphere. This sudden discharge caused so great a depression in the cylinder that the free piston rose and came to rest at about two-thirds of its upward travel. With photo-electric apparatus, Davies recorded the piston travel and showed that the entire process was completed in a very short time. With a cylinder of 2·4 in. bore and 2·2 in. effective stroke, the exhaust process lasted only 3·2 milliseconds, and the residual depression was 17 in. of mercury absolute. Experiments with various lengths of exhaust pipes showed that the resulting depression and the time during which the exhaust slot remained uncovered were largely unaffected by the exhaust pipe; for instance, when the exhaust-pipe length was changed from 7·5 in. to 53 in. the duration of the exhaust period changed only about 6 per cent. The gas flow in the exhaust pipe was shown by high-speed photography of a light-weight "cursor" in a glass exhaust pipe. This revealed a very rapid back-and-forth movement of the gas column.

A Junkers opposed-piston engine of 2·56 in. bore and 8·27 in. stroke was converted to the Kadenacy system, and pressure records taken at 650 r.p.m. from the inlet and exhaust ducts by a cathode-ray indicator gave the results shown in Fig. 1 by curves *a* and *b*, respectively. It will be seen that the pressure changes are transmitted from the exhaust ducts to the intake ducts without appreciable delay. The instants of opening and closing of the exhaust are shown by points *p* and *q* on curve *b*, while the corresponding points for the inlet are *s* and *t* on curve *a*. Tests at 800 r.p.m. with pipes *a*, *b* and *c* of the lengths and diameters indicated in Fig. 2 gave the curves bearing the corresponding letters. It can be seen that the point of maximum depression is retarded when the exhaust-pipe length is increased. Points *p* and *s* on the base line indicate the opening of the exhaust and inlet, respectively.

Conversion of the Junkers engine to the Kadenacy system resulted in a substantial increase of power. Test results published in November, 1939, in *Gas and Oil Power* are reproduced as curves in Fig. 3, page 242, in which the b.m.e.p. before and after

conversion is shown by curves a and b. Corresponding brake horse-power values are shown in curves c and d. The maximum power was raised from about 11 h.p. to 25 h.p., an increase of 130 per cent. This is claimed to have been obtained without any alteration to the combustion chamber or fuel-injection equipment, solely by changing the characteristics of the inlet and exhaust ports and passages, in accordance with the Kadenacy patents. The scavenge pump was rendered inoperative, and the inlet ports, which were still controlled by the upper piston, were arranged so that they communicated directly with the atmosphere. With the Kadenacy system fitted, it is claimed, it was possible to run the engine to a much higher speed than the standard engine without any ill effects. The pistons remained in a cooler condition, the maximum pressures were lower, and the specific fuel consumption also was lower.

Fig. 2.

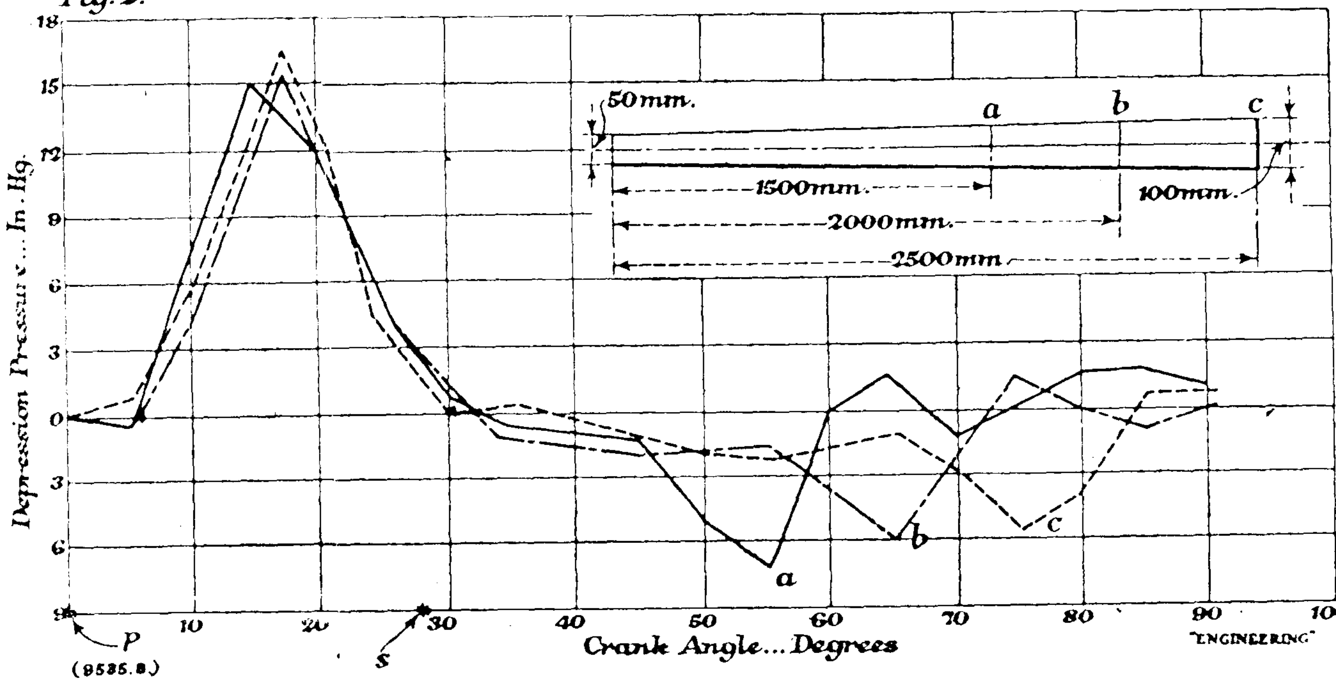


Fig. 4, page 242, shows the cylinder pressures in the Kadenacy-Junkers engine when operating without the scavenging-air pump. It is notable that a depression of over 1 lb. per square inch below atmospheric pressure occurred in the cylinder 50 deg. after the exhaust ports had opened. The letters p and q indicate the opening and closing of the exhaust while the letters s and t indicate the opening and closing of the inlet.

Conversions of other two-stroke engines, of either the uniflow or the cross-scavenge type, gave similar results, although not quite such favourable ones as those shown in Fig. 3. Some engines fitted with the Kadenacy system had blowers, others had none. When a blower was used, the scavenge-air pressure was considerably less than for a normal engine and it decreased when the load increased. In the original engine, the scavenging pressure increased with the load. The difference is explained by the fact that, with the Kadenacy system, as the load becomes greater the energy in the exhaust also becomes greater and with it the suction effect in the cylinder, thus reducing the resistance to the delivery of air from the blower. Converted engines consistently showed appreciable power increase with lower specific fuel consumption and lower exhaust temperatures. One test series carried out with different lengths of tail pipe (from 0 to 150 ft.) beyond the silencer showed that the superior performance of the engine was practically independent of the exhaust pipe length.

The results obtained with the Kadenacy system are fully discussed in the literature on the subject, but the constructional details are not disclosed. One or more of the expedients shown in the patent drawings might have been employed to obtain the excellent results quoted, but proper port timing was undoubtedly a most important factor in every case.

Kadenacy claims that, upon the opening of the exhaust port, the gas leaving the cylinder with a very high velocity will evacuate it and leave a partial or complete void behind, irrespective of whether a pipe is attached to the cylinder or not. In order to measure the depression following a sudden outflow of gas, special experimental equip-

ment has been built at the Pennsylvania State College. It consists of the steel cylinder *a*, Fig. 5, page 242, closed by a lid *b* locked with a quick acting latch *c*. This lid is forced open by the initial cylinder pressure, when the latch is released. Orifices, such as *o*, of different diameters, or nozzles, were fixed at the lid end of the cylinder. A pick-up, for determining the lowest pressure reached in the cylinder during the exhaust, was fixed at the closed end of the cylinder. This pick-up consists of a brass diaphragm *d*, 2¼ in. in diameter and 0·025 in. thick, exposed to the cylinder pressure on one side and to an adjustable depression in the space *e* on the other side. The diaphragm opens an electric contact when the pressure in the cylinder *a* drops below the adjusted depression in *e*; the contact is inserted in a 6-volt circuit in series with a transformer coil *f*. A neon lamp *g* is placed in series with the secondary winding *h* of the transformer; this gives a flash when the primary circuit opens.

The depressions have been measured for different orifice diameters and different initial tank pressures. The results are shown in Fig. 6, page 242, where the maximum depression after the gases have rushed out of the cylinder are plotted against the cylinder gauge-pressure before the cylinder was opened. Curves *a*, *b*, *c*, *d*, and *e* correspond to the use of a thin-plate orifice of 2 in., 3 in., 4 in., 5 in. and 6 in. diameter, respectively. With the 6-in. orifice there was, in effect, no plate, as the cylinder was wide open. Curve *f* corresponds to the use of a convergent-divergent nozzle, 3⅛ in. in length, with a throat diameter of 2 in. The opening on the pressure side was 4½ in. and convergence took place for 2 in. Over the remaining 1⅛ in. the nozzle diverged uniformly, its sides making an angle of 15 deg. with the nozzle axis. The convergence to the throat was smooth and rounded. The experiments show that the depression is great enough to account for the results on two-stroke Diesel engines applying Kadenacy's patents.

Various explanations of the Kadenacy effect have been published. None of them gives as close a concordance with our experimental results as Geyer's theory, which was published in ENGINEERING, vol. 151, page 463 (1941). Geyer assumes that the potential energy of the compressed air contained in a cylinder of volume V is transformed into kinetic energy and into work against atmospheric pressure. At the instant when the pressure in the cylinder drops to that of the atmosphere, the kinetic energy of the air still contained in the cylinder will perform work against the atmospheric

Fig. 8.

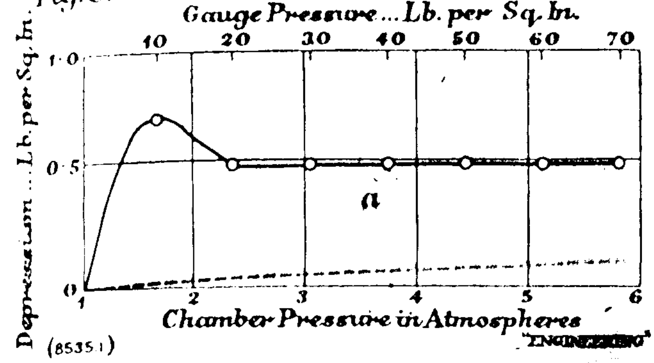


Fig. 9.

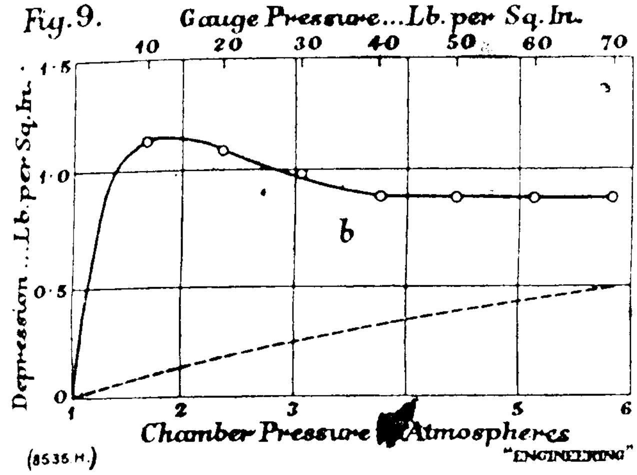
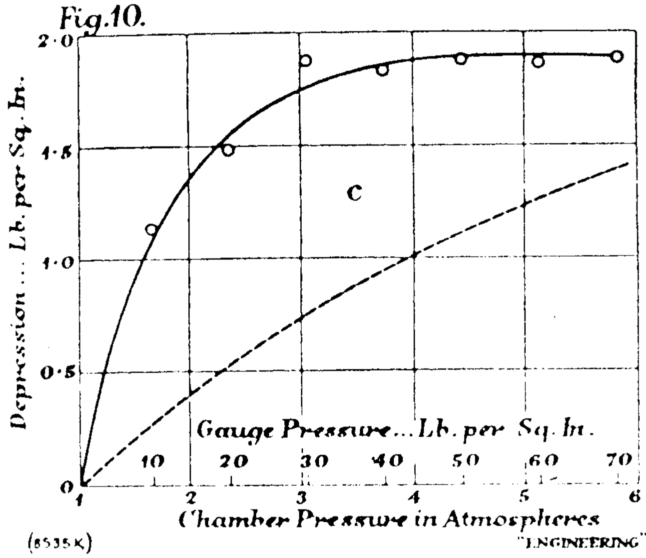


Fig. 10.



Even when the pressures in the vessel have become equal to the atmospheric pressure, further expansion is possible, due to the velocity of the gases contained in the vessel and in the nozzle.

Assume a nozzle to be interposed between the cylinder and the outside atmosphere, and let *m* V represent the volume contained in the nozzle; then the work which can be done against the atmospheric pressure p_0 is :—

$$\frac{1}{3} W_1 \frac{u_1^2}{2g} + m W_1 \frac{u_2^2}{2g} = \frac{u_1^2}{2g}\left(\frac{1}{3} + a^2 m\right) W_1.$$

If V_1 is the volume of gases expanded beyond the orifice, the work is :—

$$p_0 V_1 = \frac{u_1^2}{2g}\left(\frac{1}{3} + a^2 m\right) W_1 \quad . \quad . \quad (2)$$

Assuming the expansion is adiabatic,

$$p_0 (V + m V)^k = p (V + m V + V_1)^k$$

$$p = \frac{p_0}{\left[1 + \frac{V_1}{V(1+m)}\right]^k} \quad . \quad . \quad (3)$$

Let

$$\frac{u_1^2}{2g} \cdot \frac{X^{\frac{k-1}{k}}}{R T_1} = \frac{\frac{J C_v}{R}\left(X - X^{\frac{1}{k}}\right) - \left(X^{\frac{1}{k}} - 1\right)}{\frac{1}{3} + a^2 \left(X^{\frac{1}{k}} - 1\right)} = K$$

so that equation (2), namely :—

$$p_0 V_1 = \frac{u_1^2}{2g}\left(\frac{1}{3} + a^2 m\right) \frac{V p_0}{R T_1} X^{\frac{k-1}{k}}$$

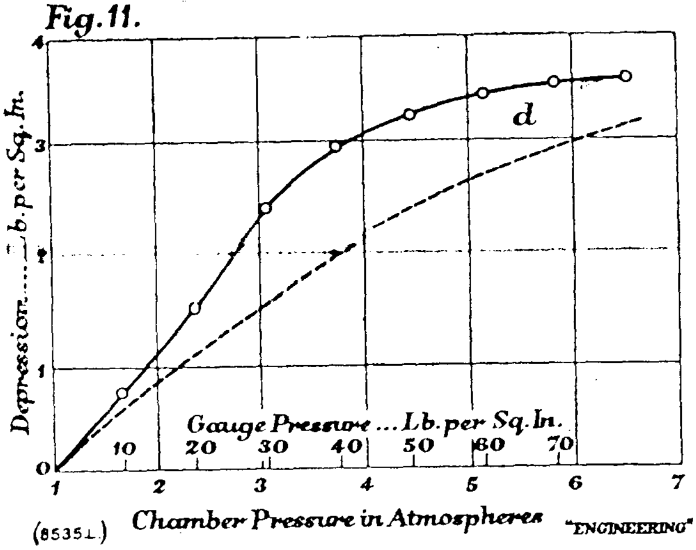
may be written :—

$$\frac{V_1}{V} = K\left(\frac{1}{3} + a^2 m\right),$$

whence the resulting pressure in the cylinder is

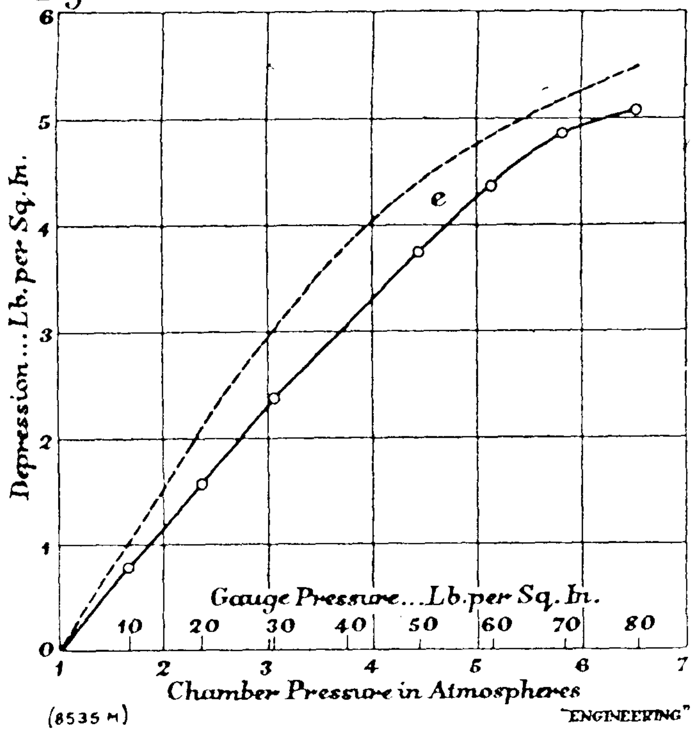
$$p = \frac{p_0}{\left(1 + \frac{\frac{1}{3} + a^2 m}{1 + m} k\right)^k} \quad . \quad . \quad (4)$$

Fig. 11.



This theory, when applied to the apparatus shown in Fig. 5, gives the depression values $p_0 - p$ shown in Fig. 7, where, as in Fig. 6, a, b, c, d, e and f are curves for 2 in., 3 in., 4 in., 5 in., 6 in. orifices, and a 2-in. nozzle, respectively. Comparison between theoretical and experimental results is made in Figs. 8 to 13. In these, the curves are

Fig. 12.



designated by the same letters are are used in Figs. 6 and 7, and distinction between them is facilitated by drawing the theoretical results in dotted lines. It is seen that, for a fully open end, the calculated depressions agree with the observed values, but for small orifices the observed depressions are much greater than those calculated. The neglected density change may account for this.

Geyer's theory may be used for calculating the discharge velocities of the air from the vessel. The velocity of the air outflow u_2 is calculated from equation (1) with the relation $u_2 = a\, u_1$ and the results are given in Fig. 14 by the curves against which is written $a = 1$, applying to the open cylinder, and $a = \infty$, applying to zero orifice area. The sound velocities calculated by the formula $V = \sqrt{32 \cdot 2\, k\, R\, T}$ are also indicated as short horizontal lines, for various values of T_1 and taking k as 1·4. The values of T have been reduced to deg. F. and written against the appropriate sound velocities.

For each value of the pressure ratio X the velocity u_2 is between the value calculated for an entirely open tank ($a = 1$) and the limit when a tends to infinity. These two curves lie close enough together to permit of estimating the outflow velocity for any initial cylinder pressure, independently of the exhaust port area. It will be noted that for $X = 5$, this velocity happens to equal that of the sound in the compressed gas. $X = 5$ and 14·5 lb. per square inch ambient pressure corresponds to a

cylinder pressure of 72 lb. per square inch, which, incidentally, about represents the normal conditions in an engine when the exhaust port opens. The outflow velocities do not seem to differ much from the sound velocities.

In conclusion, it may be stated that the agreement between Geyer's theory and our experimental

Fig. 13.

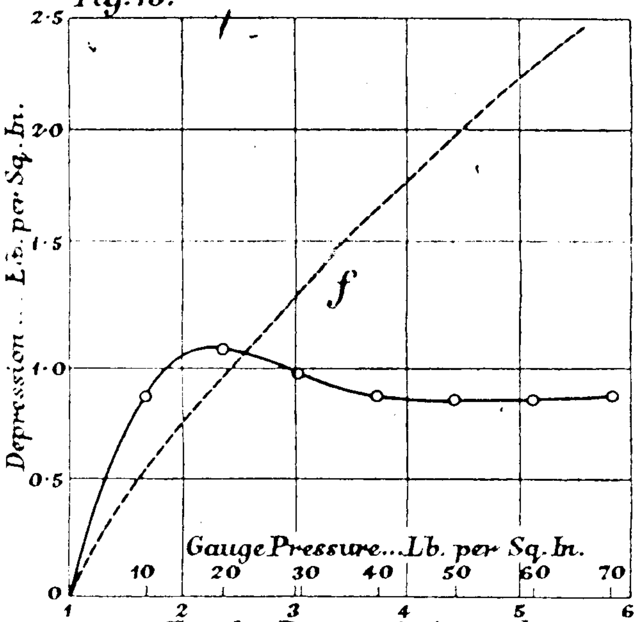


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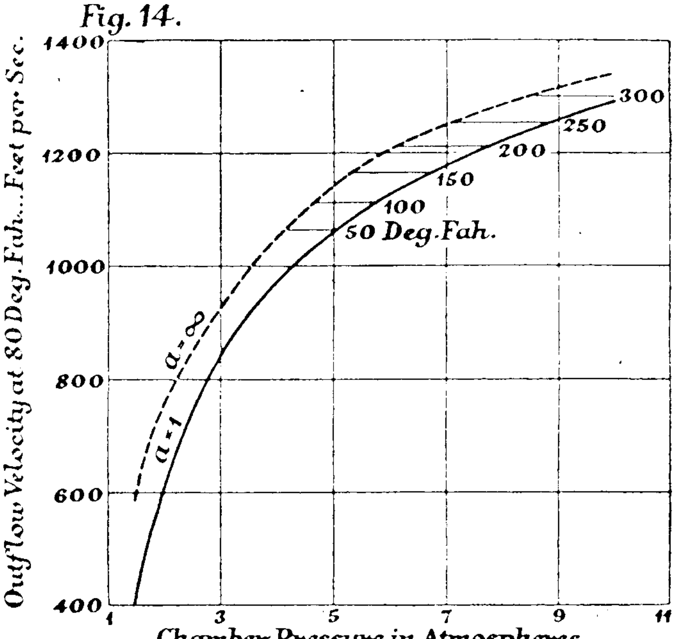
Fig. 14.



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results is good in the case of large exhaust-orifice areas, but poor when the restriction is considerable. Some of Geyer's assumptions are not justified, and more realistic assumptions would, no doubt, change the results to some degree. Nevertheless, his theory gives a simple and apparently correct picture of the process by applying only the reasoning of classical fluid mechanics. Though Geyer makes no use of the sound-velocity formula, the velocities calculated by his theory agree closely with the sound velocities which obtain under normal engine conditions.

The Kadenacy effect is sometimes described as an "implosion" that follows an explosion, by which it is meant that a sudden release of pressure from a closed vessel through a large opening is followed by a depression which causes the flow to reverse. Both experiments and theory confirm this hypothesis and there is no reason why it should not be accepted as a law of nature. Whether or not the existence of this phenomenon was suspected before, it is due to the genius of Kadenacy that this effect has been utilised for charging two-stroke cycle engines.

A statement that recurs in most Kadenacy patents is that the burnt gases discharge through the exhaust ports at a speed much in excess of the speed of adiabatic expansion. Patent No. 2,123,569 (United States) gives the "speed of adiabatic expansion" as 350 m. to 450 m. per second, and gives the "ballistic speed" of exhaust due to the "ballistic force" as 1,400 m. to 1,800 m. per second, that is, about four times as high. The velocity of sound in a diatomic gas of a temperature about equal to that of the exhaust gases is, as a matter of fact, 350 m. to 450 m. per second. In a straight or convergent orifice this is also the upper limit for the discharge velocity of the gas.

The authors were unable to find out where figures of 1,400 m. to 1,800 m. per second had been obtained, but Kadenacy uses these figures in porting design and obtains good results. This may be an indirect proof that his figures are correct. Let us examine the evidence.

Kadenacy reasons that, since the pressure in the cylinder when the exhaust port opens is about 5 atmospheres, four volumes of exhaust gases have to be discharged to bring the pressure down to atmospheric during the blow-down period. It takes as much time, he states, to discharge four cylinder-volumes of gas at 1,800 m. per second velocity as one volume of gas at $\frac{1}{4} \times 1{,}800 = 450$ m. per second velocity. Therefore, he envisages the "hypothetical velocity" of 450 m. per second and designs his exhaust ports in such a manner that the gas column which projects through the exhaust port during the exhaust lead period with the hypothetical velocity of 450 m. per second is just equal to one cylinder volume. If V denotes this volume in cubic metres, A_m the mean uncovered exhaust port area during the exhaust lead (blow-down period) in square metres, and L the length of the hypothetical gas column in metres, then :—

$$V = A_m L \quad . \quad . \quad . \quad (5)$$

The hypothetical velocity, which is 450 m. per second, is given by :—

$$V_h = \frac{L}{t} \quad . \quad . \quad . \quad (6)$$

where the blow-down period in seconds is :—

$$t = \frac{\alpha}{6n} \quad . \quad . \quad . \quad (7)$$

Here α is the exhaust lead in degrees, and n is the engine speed in revolutions per minute.

From (5), (6) and (7) :—

$$V = A_m V_h \frac{\alpha}{6n} \quad . \quad . \quad (8)$$

Putting $V_h = 450$, and re-arranging, we get :—

$$A_m \alpha = V n \frac{6}{450} = \frac{Vn}{75} \quad . \quad . \quad (9)$$

This is Kadenacy's reasoning and the resulting formula is identical with that of Kadenacy's U.S. Patent No. 2,144,065, except that different symbols are used. From this, the proper exhaust lead can be calculated. By converting the metric units used in Kadenacy's formula (9) to square inches, we get :—

$$A_m \alpha = \left(\frac{39 \cdot 4^2}{39 \cdot 4^3}\right) \frac{Vn}{75} = 0 \cdot 00034\ Vn. \quad . \quad (10)$$

This is still Kadenacy's formula, but in a slightly changed form. On the other hand, Schweitzer's formula (*Diesel Power and Diesel Transportation*, October, 1942, page 818), Ref. 6 gives :—

$$A_m \alpha = 0 \cdot 00033\ V_d\, n \quad . \quad . \quad (11)$$

where V_d is the displacement volume. Ignoring the difference between V and V_d, formula (11) is seen to differ by only 3 per cent. from formula (10).

Formula (10) is a consequence of the Kadenacy theory, but formula (11) was derived independently of the Kadenacy theory, before the writer knew of the latter's existence. The Kadenacy theory supposes the existence of "ballistic velocities" which are about four times higher than those obtained from adiabatic expansion. Formula (11) was derived by conventional thermodynamics, and the discharge velocities used were substantially the same as sound velocities, modified by discharge coefficients obtained by the Nusselt theory. Thus the two theories give practically identical results.

We believe Kadenacy's reasoning to be fallacious and in the following we try to show the error in it :—

Kadenacy states (Patent No. 2,123,569, page 4, lines 33-62) that instead of applying the true ballistic velocity, 1,800 m. per second, to four cylinder-volumes of gases to bring down the pressure from 5 atm. to 1 atm. one may use a hypothetical velocity of 1,800 ÷ 4 = 450 m. per second to one cylinder volume and get the same results. We believe this to be a mistake. One cylinder volume should be considered, and not four, even if the pressure in the cylinder is 5 atm.

A rough reasoning should suffice to make this clear. Let us consider two cases. In the first case, the density of the gas does not vary during the discharge. Before, in, and after the exhaust orifice the density is the same. In this case (which is analogous to the discharge of water) if the cylinder volume V contains gas at 5 atm. pressure and that is discharged through an orifice A with 450 m. per second discharge velocity, and $A = \frac{V}{450}$, then at the end of one second all of the gas will be discharged and not only one-fifth of it. It makes no difference whether the pressure in the cylinder was 5, 4, 3, 2 or 1 atm. In every case 450 m. per sec. discharge velocity is required to evacuate the cylinder in one second. The assumption of a ballistic velocity of 1,800 m. per second is unjustified.

Let us next consider the change in gas density. The discharge velocity refers to the throat of the orifice and the specific weight to be considered is the one existing in the throat.

Calculation by conventional thermodynamics shows that the specific weight in the throat is approximately 0·78 kg. per cubic metre at the beginning of the blow-down, and drops gradually to approximately 0·4 kg. per cubic metre at the end. Assuming a constant discharge velocity of 450 m. per second and a constant exhaust orifice of A sq. m., during the t seconds of the blow-down period we shall discharge, on the average,

$$450 \times \frac{0 \cdot 78 + 0 \cdot 4}{2} \times A \text{ kg. of air per second.}$$

If, conforming to Kadency, we make $A = \frac{V}{450}$, where V is the total cylinder volume, the amount discharged will be

$$D = 0 \cdot 59\ V \text{ kg. per second.}$$

while the amount that has to be discharged to reduce the pressure from 5 atm. to 1 atm. is

$$D_1 = \frac{4}{5} V \gamma_{int} = 0 \cdot 8\ V\ 0 \cdot 73 = 0 \cdot 585\ V \text{ kg.}$$

It is seen that D is approximately equal to D_1 which means that the exhaust can be disposed of in the required time at sound velocity and that no supersonic or ballistic velocity is required. We

conclude that, while the value of Kadenacy's hypothetical velocity is correct and should give good results if applied to porting design, it in no way supports the existence of ballistic velocities in excess of sound velocities.

In trying to summarise our rather superficial investigation on the Kadenacy effect, we submit the following conclusions :

(1) The astonishing performance claimed by Kadenacy engines should not be rejected as incredible, as the effect is based on a demonstrable physical phenomenon, namely, the rarefaction that follows the sudden discharge of compressed gas from a closed vessel.

(2) The Kadenacy effect is not due solely to the inertia of the gas column in the exhaust pipe, as 10 in. Hg. depression was observed by us in a vessel without any exhaust pipe, and reported results show good performance in engines with varying lengths of exhaust pipes.

(3) The Kadenacy effect vanishes if the discharge opening becomes relatively small. One inference is that full advantage cannot be taken of the Kadenacy effect because exhaust ports or valves cannot be opened fast enough.

(4) Geyer's theory seems to give a fair picture of the mechanics of the Kadenacy effect, and excellent numerical agreement in case of large discharge openings.

(5) Nothing in our observations supports Kadenacy's contention that velocities in excess of sound velocities are involved in the exhaust process. Conventional thermodynamics based on simple adiabatic expansion gave formulæ practically identical with those recommended by Kadenacy, and Geyer's theory applied to our experiments also indicated discharge velocities of the order of sound velocities and no higher.

ASME Transactions vol. 68 10/46 pp. 729-34

Taking the Mystery Out of the Kadenacy System of Scavenging Diesel Engines

By P. H. SCHWEITZER,[1] C. W. VAN OVERBEKE,[2] and L. MANSON[3]

The "Kadenacy effect" of the Diesel-engine exhaust is utilized to create a vacuum in the cylinder for introducing the fresh charge. The result of the application of this system is exemplified in tests in 1939, of a converted Junkers opposed-piston engine in which the power was raised from 11 to 25 hp, or an increase of 130 per cent. This was accomplished solely by changing the characteristics of the inlet and exhaust ports and passages in accordance with the Kadenacy patents. However, the theory expounded by the inventor has been subject to question, as the authors explain, and while the results are entirely matters of record, the phenomenon can be accounted for by conventional thermodynamics based upon simple adiabatic expansion. This fact the authors have demonstrated by experiment. The formulas derived check closely with those of Kadenacy but they are based upon a more rational approach to the problem than Kadenacy's contention that supersonic velocities are involved in the exhaust process.

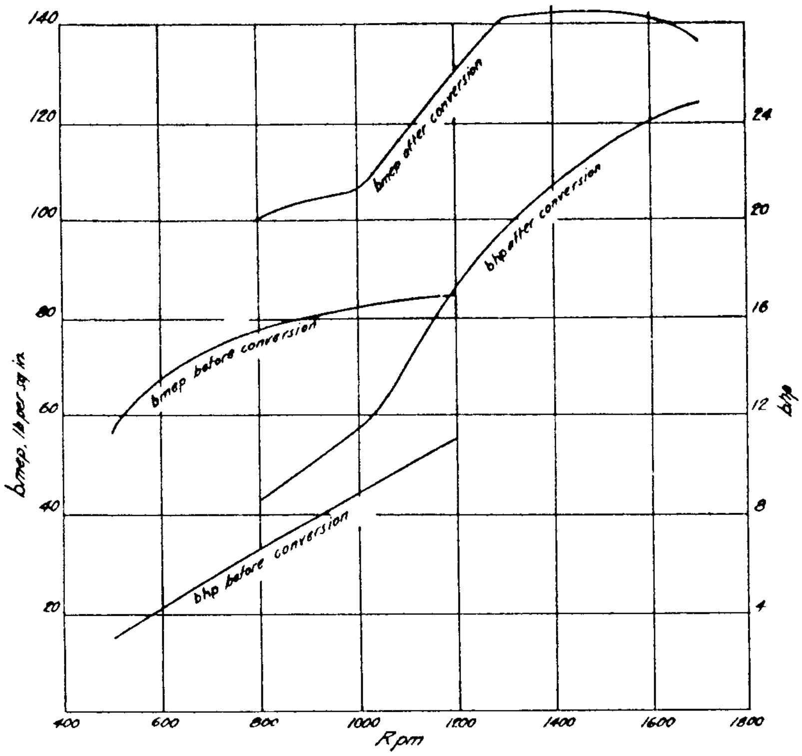

Fig. 1 Horsepower and Brake Mean Effective Pressure of Junkers Opposed-Piston Engine Before and After Conversion to Kadenacy System

During the last few years the Kadenacy system of scavenging Diesel engines has attracted some attention in this country and more abroad. That system is best known in the form of a blowerless two-stroke-cycle engine. The "Kadenacy effect" of the exhaust is utilized to create a vacuum in the cylinder for introducing the fresh charge. That by itself would not be remarkable if we were to attribute the effect to a tuned exhaust pipe. The Kadenacy system, however, embraces more than that. In numerous patents (1)[4] an ever-recurring sentence is that "at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when a flow resulting from an adiabatic expansion only is involved, and in such a short interval of time that it is discharged as a mass, leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder, etc." This has been quoted from U. S. Patent 2,168,528. Kadenacy's other patents include various versions of the same statement and explain that mass means a "coherent mass" (Patent No. 2,123,569), "having properties similar to those of a resilient body" (Patent No. 2,102,559), and that the "ballistic" speeds involved in the evacuation are about 4 times higher than "the speed of adiabatic expansion," leaving behind them "a high depression which may reach a complete vacuum" (Patent No. 2,131,957).

The foregoing notions are so unorthodox that many engineers ignored and even ridiculed them. But it is not wise to ignore test results and Kadenacy has astonishing results to his credit.

The first commercial blowerless Kadenacy engine was built by Petter (2), and the same company is still building Kadenacy-type engines of an improved design (3); however, with a blower attached.

According to tests (4), described in 1939, the conversion of a Junkers opposed-piston engine resulted in a substantial increase of power. Some test results are reproduced in Fig. 1. The maximum power was raised from about 11 to 25 hp, an increase of 130 per cent. This is claimed to have been obtained without any alteration to the combustion chamber or fuel-injection equipment, solely by changing the characteristics of the inlet and exhaust ports and passages, in accordance with the Kadenacy patents. The scavenge pump was rendered inoperative and the inlet ports, which were still controlled by the upper piston, were arranged so that they communicated directly with the atmosphere. With the Kadenacy system fitted to the engine it was possible to run the engine to a much higher speed without any ill effects, it is claimed. The pistons remained in a cooler condition, the maxi-

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[2] Ex-Cell-O Corporation, Detroit, Mich.

[3] DeLaval Steam Turbine Company, Trenton, N. J.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 26-29, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

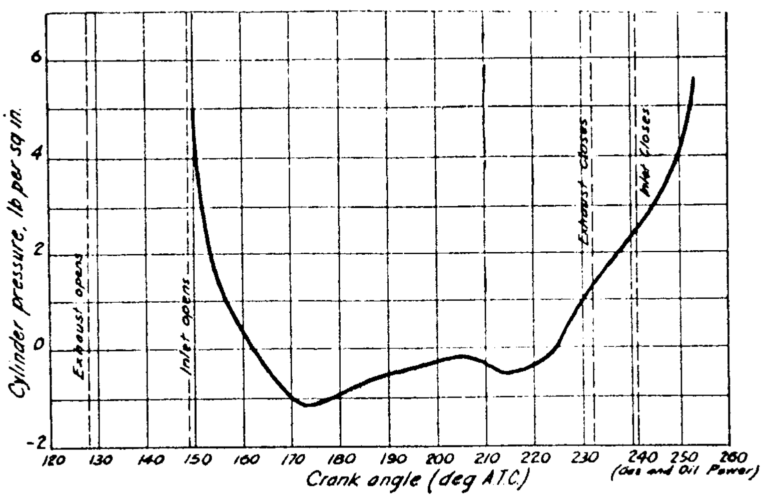

Fig. 2 Cylinder Pressures in the Junkers-Kadenacy Opposed-Piston Engine

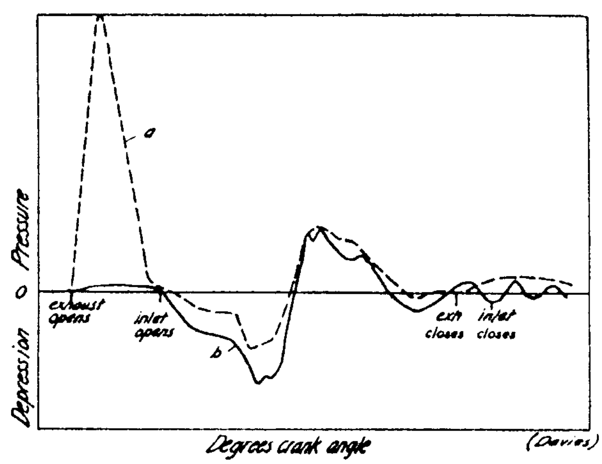

Fig. 3 Pressure Fluctuations in Exhaust Duct—Line A—and Inlet Duct—Line B—of a Junkers-Kadenacy Engine; 650 Rpm

mum pressures were lower and the specific fuel consumption also was lower.

Fig. 2 shows the cylinder pressures in the Kadenacy-Junkers engine operating without scavenging air pump. It is notable that a depression over 1 psi below atmospheric occurred in the cylinder 50 deg after the exhaust ports had opened.

Fig. 3 shows the pressures in the intake and exhaust ducts of the converted Junkers engine from which it is seen that pressure changes are transmitted from the exhaust ducts to the intake ducts without appreciable delay. Tests at 800 rpm with various lengths of exhaust pipes gave Fig. 4 which shows that the Kadenacy effect does not depend upon a particular pipe length, only the time of maximum depression was retarded when the exhaust-pipe length was increased.

We have dwelt on this Kadenacy-Junkers engine because part of the tests reported were made by such an unquestioned authority as Prof. S. J. Davies of London, England.

Conversions of other two-stroke engines of either uniflow or the cross-scavenge type gave similar results, although not quite as spectacular as those shown in Fig. 1. Some engines fitted with the Kadenacy system had blowers and others had none. When a blower was used the scavenge-air pressure was considerably less, and significantly it decreased when the load increased. In the original engine the scavenging pressure increased with the load. The difference is explained by the fact that with the Kadenacy system as the load becomes greater the energy in the exhaust also becomes greater and with it the suction effect in the cylinder, thus reducing the resistance to the delivery of the air from the blower. Converted engines consistently showed appreciable power increase with lower specific fuel consumption and lower exhaust temperatures.

The mechanics of the Kadenacy principle were investigated by Davies (5) with a free-moving piston in a cylinder. He compressed a mixture of air and gasoline vapor with a hand crank and ignited it with a spark plug. The gas pressure sent the piston downward until it uncovered a slot through which the burnt gases discharged into the atmosphere. This sudden discharge caused so great a depression in the cylinder that the free piston rose in the cylinder and came to rest at about two thirds of its upward travel. With photoelectric apparatus, Davies recorded the piston travel and showed that the entire process was completed in a very short time. With a cylinder of 2.4 in. bore and 2.2 in. effective stroke the exhaust process lasted only 3.2 milliseconds and the residual depression was 17 in. Hg abs. Experiments with various lengths of exhaust pipes showed that the resulting depression and the time during which the exhaust slot remained uncovered were largely unaffected by the exhaust pipe. For instance, when the exhaust-pipe length was changed from 7.5 to 53 in., the duration of the exhaust period changed only about 6 per cent. The gas flow in the exhaust pipe was shown by high-speed photography of a lightweight "cursor" in a glass exhaust pipe. This revealed a very rapid back-and-forth movement of the gas column.

While the results obtained with the Kadenacy system are fully discussed in the literature, the constructional details are not disclosed. One or more of the expedients shown in the patent drawings such as tapered exhaust pipes and reflection-wave stoppers might have been employed to obtain the excellent results but apparently no particular importance is attached to them.

In view of these reports, the authors felt a necessity to de-

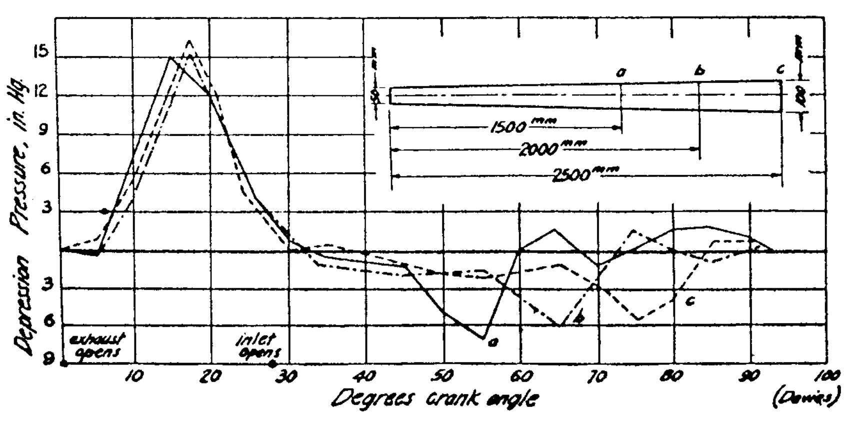

Fig. 4 Pressures in Exhaust Pipe of Junkers-Kadenacy Engine With Various Lengths of Exhaust Pipes; 800 Rpm

termine in a conclusive manner whether the so-called Kadenacy effect really exists, aside from the pipe effect the nature of which is fairly well known (6).

Investigation of the Kadenacy Effect

Kadenacy claims that upon opening the exhaust port the gas leaving the cylinder with a very high (supersonic) velocity will evacuate it, leaving a partial or complete void behind, irrespective of whether a pipe is attached to the cylinder or not.

The depression thus created in a cylinder, no matter how short in duration, can be measured and no combustion needs to be involved in the test.

In order to measure the depression following a sudden gas outflow, a special setup was built at The Pennsylvania State College. It consists of a steel cylinder, Fig. 5, closed by a lid locked with a quick-acting latch. This lid is forced open by the initial cylinder pressure when the latch is released. Orifices of different diameters or nozzles were fixed at the lid end of the cylinder. A pickup for determining the lowest pressure reached in the cylinder during the exhaust was fixed at the closed end of the cylinder. This pickup consists of a brass diaphragm of $2^1/_4$ in. diam and 0.025 in. thick, exposed to the cylinder pressure on one side, and to an adjustable depression on the other. The diaphragm opens an electric contact when the pressure in the cylinder drops below the adjusted depression; the contact is inserted in a 6-volt circuit in series with a transformer coil. A neon lamp is placed in series with the secondary winding of the transformer. This gives a flash when the primary circuit opens.

The depressions have been measured for different orifice diameters and different initial tank pressures. The results are shown in Fig. 6 where the maximum depression after the gases have rushed out of the cylinder, are plotted against the cylinder gage pressure before the cylinder was opened. With the 6-in. orifice there was in effect no plate, as the cylinder was wide open. One curve corresponds to the use of a convergent-divergent nozzle with 2-in. throat diameter and $3^1/_2$-in. length. The opening on the pressure side was $4^1/_2$ in. and convergence took place for 2 in. Over the remaining $1^1/_2$ in. the nozzle diverged uniformly at an angle of 15 deg.

The experiments show that the depression, although far from a complete void, is great enough to account for the results on two-stroke Diesel engines applying Kadenacy's patents.

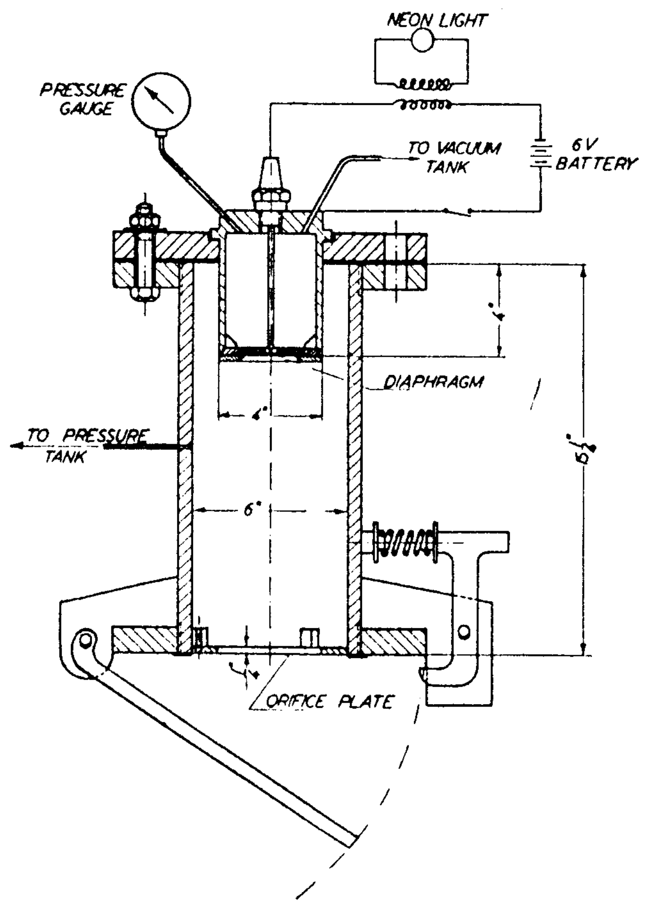

Fig. 5 Vessel With Quick-Opening Lid to Test Kadenacy Effect

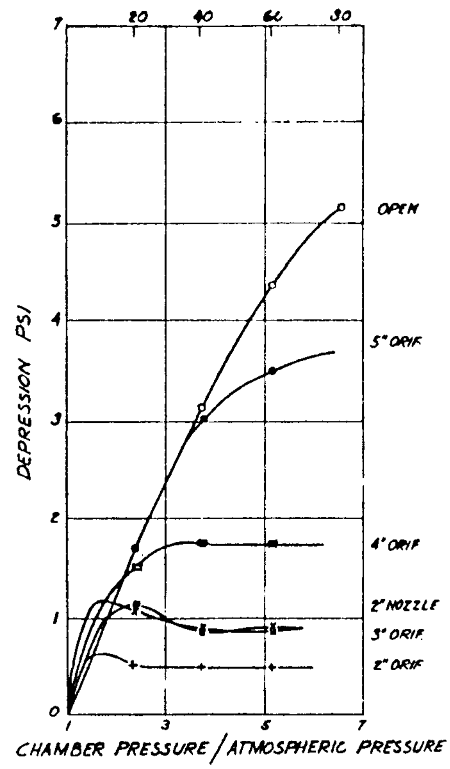

Fig. 6 Kadenacy Effect; Observed Depressions by Sudden Pressure Release, Penn State Experiments

Theory of the Kadenacy Effect

Various explanations of the Kadenacy effect have been published. Giffen (7) calculated the pressure waves generated by the sudden evacuation of a cylindrical vessel and obtained depressions of the order that we have observed. His calculations even showed that with an orifice appreciably smaller than the cylinder the depression does not increase continuously with the increase of the initial cylinder pressure but it reaches a maximum when the initial cylinder pressure is around 20 psia, and if it exceeds that, the depression begins to decrease. This agrees with our observations with orifices and nozzles of 3 in. diam or smaller.

Geyer (8) proposed a rather simple theory of the Kadenacy effect. He attributes the evacuation of the cylinder to the kinetic energy of the gases still in the cylinder, yet rushing out of the cylinder.

It is not improbable that Geyer got the idea for his theory from Kadenacy himself. For instance, in U. S. Patent No. 2,123,569, Kadenacy is using the analogy of a coil spring to explain what happens in the cylinder. He visualizes a helical spring on a table compressed, with a certain amount of energy

stored in it. If the spring is released gradually by allowing it to expand against a resistance, it will return to its free length. The work done by the spring will then become stored in the resistance. This, he states, corresponds to the release of compressed gases from a container through an orifice which is opened gradually.

On the other hand, if the spring, after having been compressed on the table, is released suddenly by removing the compressing means in a short interval of time, the spring while expanding will leave the table bodily. The energy stored in the spring imparts momentum to the spring. During its flight through the air after it has left the table, oscillations will occur in the spring but these oscillations will bear no direct relation with the motion of the spring body from the table. This case corresponds to the sudden release of the gas from the cylinder according to Kadenacy.

Geyer goes through a somewhat similar reasoning. The potential energy of the compressed gas is transformed into kinetic energy. The gas will leave the cylinder with a velocity that corresponds to this energy. At the time the pressure inside of the cylinder has dropped to atmospheric, the gas in the cylinder still has some kinetic energy, which will perform work against the atmospheric pressure. This work consists of displacing a certain volume of air against ambient pressure. The equivalent volume of air to replace the volume must come from the cylinder, and so a depression is created by the discharge of that amount of air.

Geyer's simple theory can have no pretension of describing accurately such a complex phenomenon as the sudden evacuation of a cylinder. Yet it not only gives a correct mental picture of the mechanism of the Kadenacy effect but surprisingly it even gives tolerable agreement with observed results. Geyer has applied his calculations to Davies' experiments (8), and we to the Penn State experiments (9), which are shown in Fig. 7.

In Fig. 8 is reproduced a comparison of the observed and calculated depressions when the sudden removal of the lid uncovered the full cylinder opening. With smaller orifices the observed depressions are higher than those calculated.

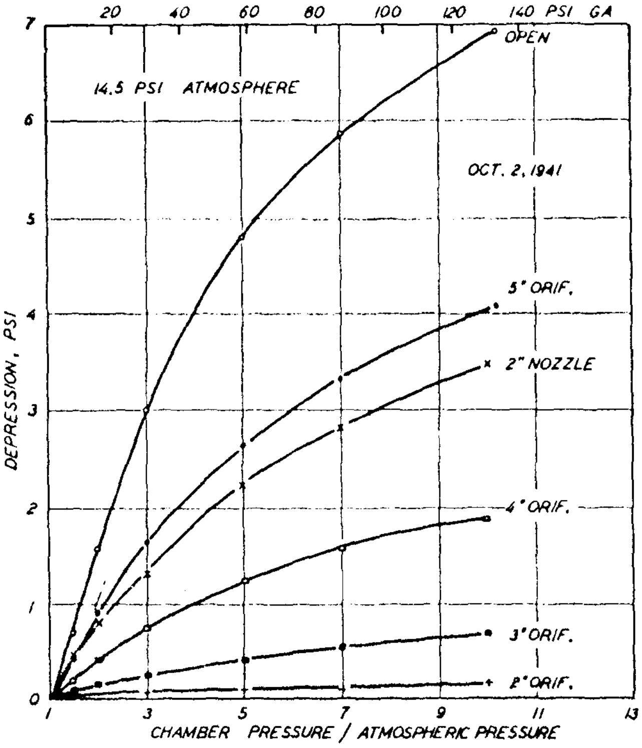

Fig. 7 Kadenacy Effect; Geyer's Theory Applied to Penn State Experiments

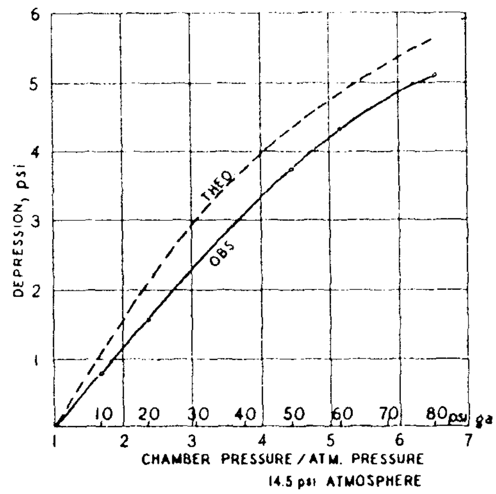

Fig. 8 Kadenacy Effect; Observed and Calculated Depressions; Open End

Ballistic Velocities

A recurrent statement in most Kadenacy patents is that the burnt gases discharge through the exhaust ports at a speed far in excess of the speed of adiabatic expansion. Patent No. 2,123,569 gives the "speed of adiabatic expansion" as 350 to 450 m per sec, and gives the "ballistic speed" of exhaust due to the "ballistic force" as 1400 to 1800 m per sec, that is about 4 times as high.

The velocity of sound in a diatomic gas of a temperature about equal to that of the exhaust gases is as a matter of fact 350 to 450 m per sec. In a straight or convergent orifice this is also the upper limit of discharge velocity of the gas.

The authors were unable to find out where figures of 1400 to 1800 m per sec had been obtained. But Kadancy uses these figures in porting design and obtains good results. This may be an indirect proof that his figures are correct. Let us examine this evidence.

Since the pressure in the cylinder when the exhaust port opens is about 5 atm, Kadenacy reasons, 4 volumes of exhaust gases have to be discharged to bring the pressure down to atmospheric during the blowdown period. It takes, he states, as much time to discharge 4 cylinder volumes of gas at 1800 m per sec velocity as one volume of gas at $^1/_4 \times 1800 = 450$ m per sec velocity. Therefore he envisages the "hypothetical velocity" of 450 m per sec and designs his exhaust ports in such a manner that the gas column which projects through the exhaust port during the exhaust-lead period with the hypothetical velocity of 450 m per sec, is just equal to 1 cylinder volume. If V denotes this volume in cubic meters, A_m the mean uncovered exhaust-port area during the exhaust lead (blowdown period) in square meters, and L the length of the hypothetical gas column in meters, then

$$V = A_m L \quad [1]$$

the hypothetical velocity which is 450 m per sec, is given by

$$V_{hyp} = \frac{L}{t} \quad [2]$$

where t is the time of the blowdown period in seconds, and

$$t = \frac{\alpha}{6n} \quad [3]$$

Here α is the exhaust lead in degrees and n is the engine speed in revolutions per minute.

From Equations [1], [2], and [3]

$$V = A_m V_{hyp} \frac{\alpha}{6n} \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

Putting $V_{hyp} = 450$ and rearranging, we get

$$A_m \alpha = V n \frac{6}{450} = \frac{V n}{75} \quad m^2 \deg \ldots\ldots\ldots [5]$$

This is Kadenacy's reasoning, and the resulting formula is identical with Kadenacy's formula Patent No. 2,144,065

$$\frac{W}{100\,KA} = \frac{a}{360\,N} = t$$

except for the different symbols. From this the proper exhaust lead can be calculated. By converting the metric units used in Kadenacy's Equation [5] to square inches we get

$$A_m \alpha = \frac{(39.4)^2}{(39.4)^3} \frac{Vn}{75} = 0.00034\, Vn \ldots\ldots\ldots [6]$$

This is still Kadenacy's formula in a slightly changed form. On the other hand Schweitzer's formula (10) gives

$$A_m \alpha = 0.00033\, V_{disp} n \ldots\ldots\ldots\ldots\ldots [7]$$

Ignoring the difference between V and V_{disp}, Equation [7] differs only 3 per cent from Equation [6].

Equation [6] is a consequence of the Kadenacy theory, but Equation [7] was derived independently of the Kadenacy theory, before we knew of the latter's existence. The Kadenacy theory supposes the existence of "ballistic velocities" which are about 4 times higher than those obtained from adiabatic expansion. Equation [7] was derived by conventional thermodynamics, and the discharge velocities used were substantially the same as sound velocities modified by discharge coefficients obtained by the Nusselt theory. The two theories give practically identical results.

The circumstance that Kadenacy got correct results from incorrect premises casts a doubt on Kadenacy's reasoning.

We believe Kadenacy's reasoning to be fallacious and in the following we try to show the error in it.

Fallacy in Kadenacy Theory

Kadenacy states[6] that instead of applying the true ballistic velocity, 1800 m per sec to 4 cylinder volumes of gases to bring down the pressure from 5 atm to 1 atm, one may use a hypothetical velocity of 1800/4 = 450 m per sec to 1 cylinder volume and get the same results. We believe this to be a mistake. One cylinder volume should be considered and not 4, even if the pressure in the cylinder is 5 atm.

A rough reasoning should suffice to make this clear. Let us consider two cases. In one case the density of the gas does not vary during the discharge. Before, in, and after the exhaust orifice, the density is the same. In this case (which is analogous to the discharge of water) if the cylinder volume V contains gas at 5 atm pressure and that is discharged through an orifice A with 450 m per sec discharge velocity, and $A = V/450$, then at the end of 1 sec all of the gas will be discharged and not $^1/_5$ of it only. It makes no difference whether the pressure in the cylinder was 5, 4, 3, 2, or 1 atm. In every case 450 m per sec discharge velocity is required to evacuate the cylinder in 1 sec. The assumption of a ballistic velocity of 1800 m per sec is unjustified.

[6] U. S. Patent No. 2,123,569, p. 4, lines 33–62.

Let us next consider the change in gas density. The discharge velocity refers to the throat of the orifice, and the specific weight to be considered is the one existing in the throat.

Calculation by conventional thermodynamics shows that the specific weight in the throat is approximately 0.78 kg per cu m in the beginning of the blowdown and drops gradually to approximately 0.4 kg per cu m at the end. Assuming a constant discharge velocity of 450 m per sec and a constant exhaust orifice of $A m^2$ during the t sec of the blowdown period, we shall discharge

$$450 \frac{0.78 + 0.4}{2} A \text{ kg of air}$$

If, conforming to Kadenacy, we make $A = V/450$, where V is the total cylinder volume, the amount discharged will be

$$D = 0.59\, V \text{ kg}$$

while the amount that has to be discharged to drop the pressure from 5 atm to 1 atm is

$$D_1 = \frac{4}{5} V \gamma_{int} = 0.8\, V\, 0.73 = 0.585\, V \text{ kg}$$

It is seen that D approximately equals D_1, which means the exhaust can be disposed of in the required time period, assuming only sound velocity and no supersonic or ballistic velocity is required.

We conclude that while the value of Kadenacy's hypothetical velocity is sound and should give good results if applied to porting design, it in no way supports the existence of ballistic velocities in excess of sound velocities.

A further confirmation of this conclusion can be found in calculating from Geyer's theory the outflow velocities created by the sudden opening of an air-filled vessel. Fig. 9 shows these velocities, and for comparison, also the calculated sound velocities for various temperatures. For the detailed calculation the reader is

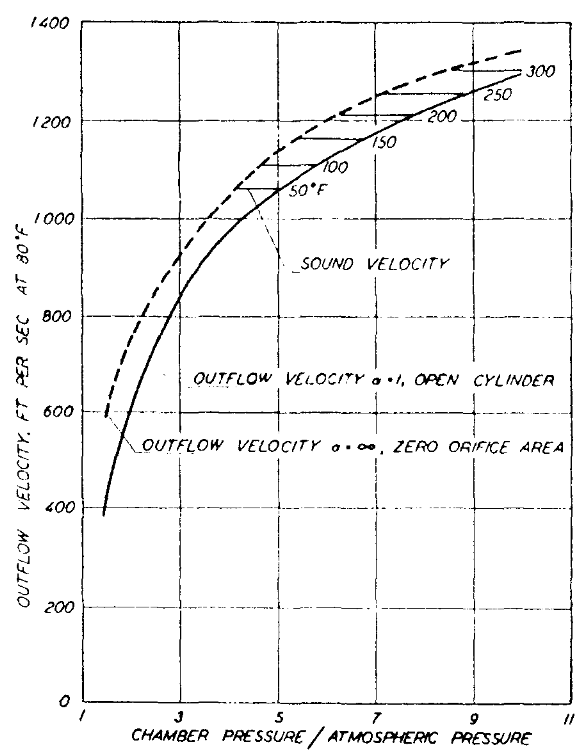

Fig. 9 Outflow Velocities Created by Sudden Opening of Vessel Filled With Air, From Geyer's Theory

referred to (9). An inspection of Fig. 9 reveals that with 5 to 7 atm chamber pressure the outflow velocities do not differ much from the sound velocities.

Conclusions

1 The astonishing performance claimed by Kadenacy engines should not be rejected as incredible as the effect is based upon a demonstrable physical phenomenon; the rarefaction that follows the sudden discharge of compressed gas from a closed vessel.

2 The Kadenacy effect is not due solely to the inertia of the gas column in the exhaust pipe, as 10 in. Hg depression was observed by the authors in a vessel without any exhaust pipe, and reported results show good performance in engines with varying lengths of exhaust pipes.

3 The Kadenacy effect vanishes if the discharge opening becomes relatively small. One inference is that an obstacle of the full exploitation of the Kadenacy effect is that exhaust ports or valves in engines cannot be opened rapidly enough.

4 Geyer's theory seems to give a fair picture of the mechanics of the Kadenacy effect and excellent numerical agreement in case of a large discharge opening.

5 Nothing in our observations supports Kadenacy's contention that velocities in excess of sound velocities are involved in the exhaust process. Conventional thermodynamics based upon simple adiabatic expansion gave formulas practically identical with those recommended by Kadenacy, and Geyer's theory applied to our experiments also successfully indicated discharge velocities of the order of sound velocities and no higher.

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Discussion

Paul Diserens.[6] The authors' discussion of the Kadenacy effect is particularly timely in view of the development of two-cycle engines, sponsored by Mr. Kadenacy, on the part of several American manufacturers at this time.

It would appear that the paper is, in effect, a continuation of the extended discussion of the Kadenacy engine which was reported in the British and the Continental technical press during the years 1938 to 1941, inclusive. While the authors make some reference to the numerous articles published at that time, it might be suggested that a bibliography comprising the more important references would constitute a valuable addition to the paper.

The writer finds it of particular interest to compare the apparatus used by the authors in their experiments, with that employed by Prof. S. J. Davies, of Kings College, University of London.[7]

It should be noted that in the Davies apparatus provision is made to insure the greatest possible speed in establishing communication between the pressure vessel and the atmosphere. In the apparatus used by the authors the speed of opening is limited because of the time required to accelerate the communicating valve.

It is evident that the results reported in the paper reflect the influence attributable to the speed of opening and therefore the results might conceivably be quite different if some other speed had been employed. Consequently, conclusion 3, i.e., "The Kadenacy effect vanishes if the discharge opening becomes relatively small." One inference that full advantage cannot be taken of the Kadenacy effect because "exhaust ports or valves cannot be opened rapidly enough," cannot be valid except when the exhaust-valve acceleration at the time of opening is low or at best comparable with that which obtained during the authors' experiments. For this reason it would appear that their results cannot properly be used as the basis for any general theory to explain the phenomena and should not be quoted as confirming such a theory based on pure speculation.

The authors assert that the hypothetical assumed average speed of 450 m per sec is derived from some preconceived theory of operation, and quote from various Kadenacy patent specifications in support of this assumption. This does not conform to my understanding of the Kadenacy patents. From a complete study of all of these it is obvious that the structures covered by the patents rest entirely upon experimental data from which empiric relationships are established. Obviously, variable coefficients must be determined in order to take care of such variable conditions as, for example, speed of opening of the exhaust valve.

An experimental engine, built by the company with which the writer is associated, develops mean effective pressures of more than 100 psi without a blower. Analysis of the exhaust gas, as well as direct measurements indicates that the volume of air passing through an engine exceeds the displacement by 60 per cent.

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[6] Director of Research and Development, Worthington Pump and Machinery Corporation, Harrison, N. J. Fellow A.S.M.E.

[7] "Sudden Discharge of Air From a Pressure Vessel," by S. J. Davies, *Engineering*, vol. 149, 1940, pp. 17–18.

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CHAPTER 13

THE KADENACY SYSTEM

13.1 The conventional picture of the scavenging process in a two-stroke cycle engine consists of first releasing the exhaust gases so that the cylinder pressure is reduced to below scavenge air pressure, then delivering the fresh air charge which is caused to sweep out most of the remaining inert gases and fill the cylinder with clean air. In section 2.3 it was stated that "In ideal scavenging the scavenge air acts like a piston in pushing the burnt gases out of the cylinder without actually mixing with them." In a certain respect an actual engine can exceed the above stated ideal. In engines employing the *Kadenacy System* the fresh air charge is not called upon to **push out** any of the exhaust gases. On the contrary, it is **drawn in** by the outrushing mass of exhaust gases or rather by the depression created in its wake.

By making use of the inertia of gas columns, a two-stroke cycle engine can even be operated without any blower, as was demonstrated by Michel Kadenacy. The first commercial blowerless engine was built in 1932 by Petter [Petter, 1934] in England, and the same company is still building Kadenacy-type engines of improved design but with a blower attached.

13.2 Penn State Tests.

The essence of the Kadenacy system is the depression created in the cylinder by a well-timed rapid opening of the exhaust. In the tests conducted by Schweitzer, Van Overbeke and Manson [Schweitzer et al., 1946] at the Pennsylvania State College, the sudden evacuation of a cylinder filled with compressed air of 80 psig pressure was followed by a vacuum of approximately 10 inches of mercury. In these experiments a vessel of 6-inch diameter and $15\frac{1}{2}$-inch length was filled with cold air of a certain pressure. Then the lid of the vessel was suddenly removed by hitting a latch as shown in Fig. 13–1 and the resulting vacuum measured by the response of a diaphragm. Kadenacy utilizes this vacuum to suck fresh air into the engine cylinder and so effect the scavenging and charging, which operations are ordinarily performed by a blower. Well-tuned exhaust (and inlet) pipes are helpful but are not the essence of the Kadenacy system. As reported by Davies [Davies, 1937] a Junkers opposed-piston engine converted to the Kadenacy system was successfully operated with various lengths of exhaust pipes from 59 to 98 inches and in a speed range from 540 to 2000 rpm. The conversion resulted in a substantial increase in power. The brake mean effective pressure was increased from 85 to 142 psi, the normal rotative speed from 1200 to 1700 rpm, and the power output from 11 to 25 hp. This is claimed to have been obtained without any alteration of the combustion chamber or fuel-injection equipment, solely by eliminating the scavenge pump and changing the characteristics of the inlet and exhaust ports and passages, in accordance with the Kadenacy patents.

13.3 Geyer's Theory.

Various explanations have been offered for the Kadenacy effect. Kadenacy himself ascribes it to a *ballistic speed* of the exhaust, said to be about four times as high as sound velocity [U. S. Patent

No. 2,131,957]. Giffen [Giffen, 1940] calculated the pressure waves generated by the sudden evacuation of a cylindrical vessel and obtained depressions of the order that have been observed. Geyer's [Geyer, 1941] rather simple theory attributes the effect to the kinetic energy of the gases rushing out of the cylinder. The potential energy of the compressed gas is, upon opening a passage, transformed into kinetic energy. The gas leaves the cylinder with a velocity that corresponds to this energy. At the time the pressure inside of the cylinder has dropped to atmospheric, most of the gas has left the cylinder, but the gas remaining in the cylinder still has some kinetic energy, which can perform work against the atmospheric pressure. This work consists of displacing a certain volume of air against ambient pressure. The equivalent volume of gas to replace that volume must come from the cylinder, and so a depression is created by the discharge of that amount of gas.

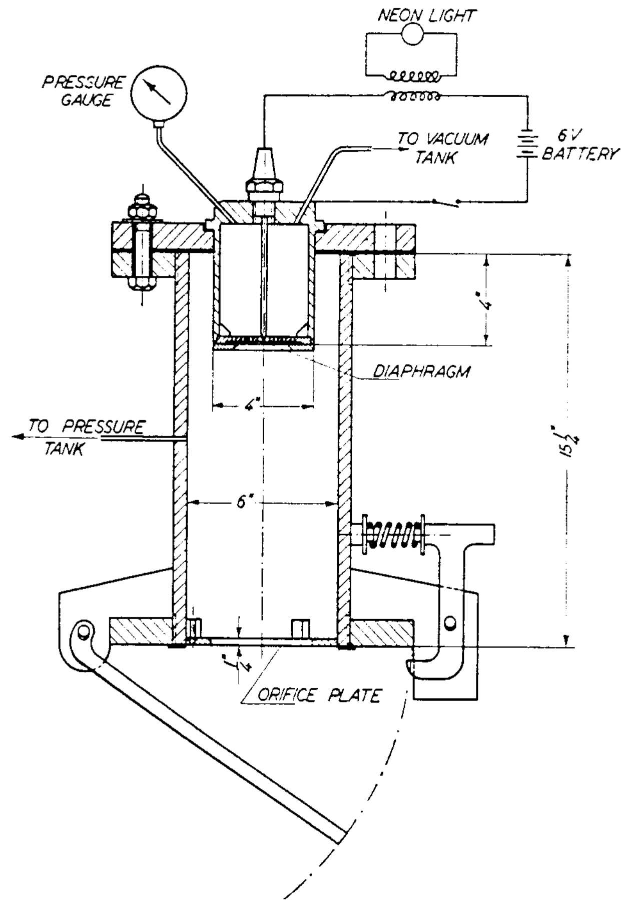

Fig. 13-1. Penn State Apparatus for Testing the Kadenacy Effect.

Geyer's simple theory can have no pretension of describing accurately a complex transient phenomenon as the sudden evacuation of a cylinder. Yet it not only gives an acceptable mental picture of the mechanism of the Kadenacy effect, but even gives tolerable agreement with observed results. Geyer has applied his calculations to Davies' experiments: Fig. 13–2 shows the comparison with the Penn State experiments, which in case of a full opening is remarkably close.

13.4 Slow Opening.

The upper two curves in Fig. 13–2 refer to the case when the entire end of the cylinder is suddenly removed. If the uncovered opening is smaller, both theory and experiment gave considerably smaller depressions. Figure 13–3 shows the resulting depressions when a 6-inch diameter cylinder containing air of 5 atmospheres pressure is evacuated by the instantaneous opening of an orifice to the atmosphere. It will be noticed that if the orifice diameter is one-half that of the cylinder (that means the discharge area one-quarter of the cylinder cross-sectional area), the resulting depression is less than 1 psi. This refers to *instantaneous* opening. Naturally, if the opening is less sudden, the resulting depression is still less. The Kadenacy effect practically vanishes if a relatively small discharge opening is uncovered relatively slowly. The inference is that the **full exploitation of the Kadenacy effect is obviated by our inability to open the exhaust ports or valves rapidly enough.**

13.5 Tuned Exhaust.

The Kadenacy effect can, however, be assisted powerfully by a properly tuned exhaust pipe. The kinetic energy of the gas contained in the exhaust pipe far exceeds the kinetic energy contained in the gas contained by the cylinder at the time of pressure equalization.

By a combination of the Kadenacy effect and a tuned exhaust pipe, the scavenging of any two-stroke cycle engine can be materially improved. Under favorable circumstances an engine can be made to run without any blower, but all engines *presently* manufactured under Kadenacy patents are built with blowers. A multicylinder engine which is to operate at variable speed does not lend itself well to blowerless operation because *with an exhaust header* the effect of tuning is less pronounced and at the wrong engine speed it is adverse. Even if a blower is used, the Kadenacy system offers decided

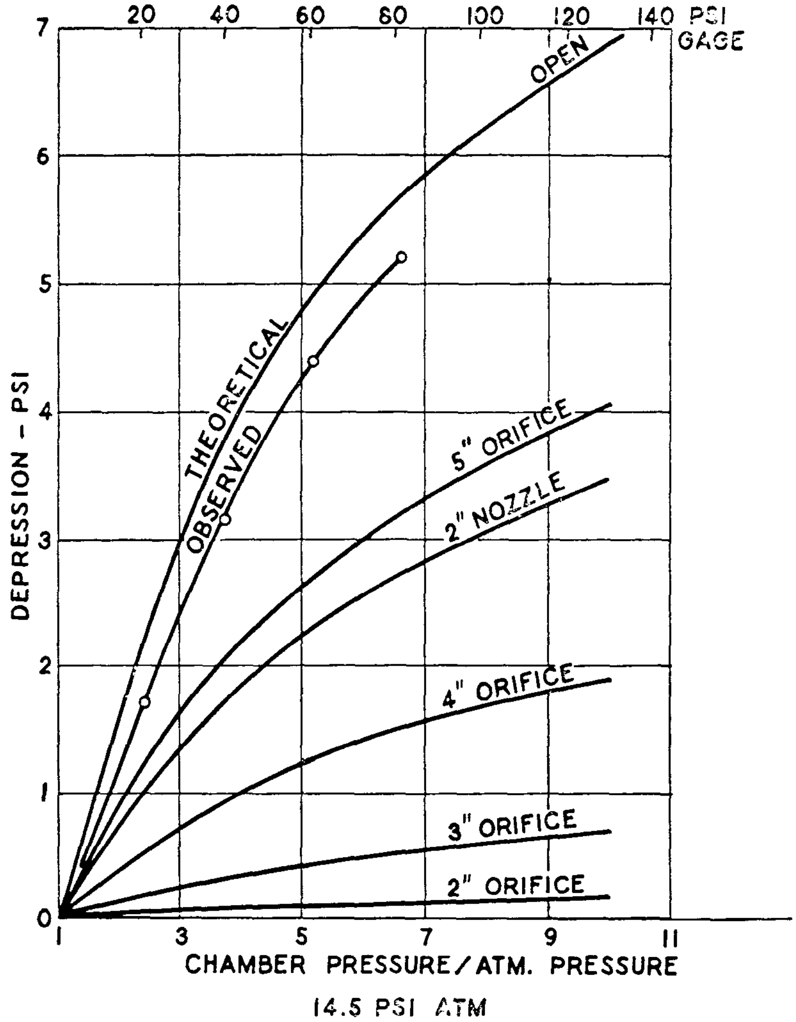

Fig. 13-2. Geyer's Theory Applied to Penn State Experiments.

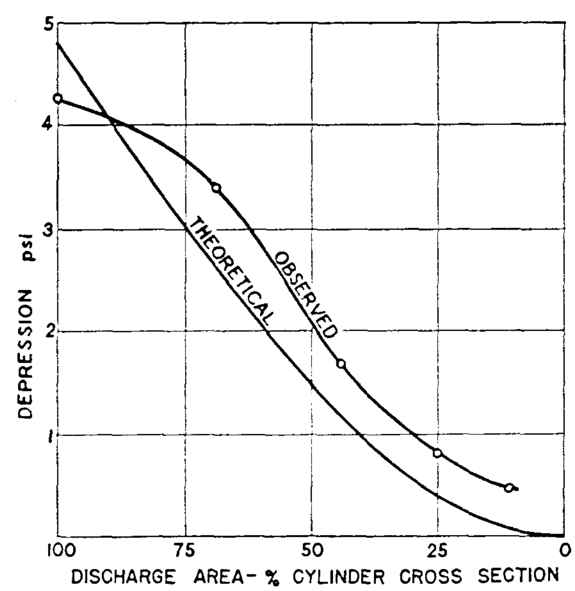

Fig. 13-3. Effect of Size of Opening on Kadenacy Effect. Chamber pressure 5 atmospheres.

advantages. The engine takes more air with a lower scavenge pressure and significantly the scavenge pressure decreases as the load increases. This is explained by the fact that as the load becomes greater the energy in the exhaust also becomes greater and by its aspiraling effect reduces the resistance to the delivery of the air from the blower. This benefits the engine in two ways: less power is used to drive the blower, and the increased air charge permits injection of more fuel with clear ex-

haust. In this way both the power output and the fuel economy are improved. Ordinarily, lower exhaust temperature also results due to the increased amount of short circuited air, and to the elimination of the temperature rise to which the air is subjected in a blower.

Although the Kadenacy effect, in the strict sense, and pipe tuning are in principle two different phenomena, they are related in nature—both are based on gas inertia—and their effects are almost indistinguishable from each other. A good way to determine whether an engine has or has not the benefit of the Kadenacy effect is to measure and plot the scavenge pressure against the engine speed while motoring and at various loads (fixed fuel rack settings). If the air-box pressure is generally less at load than at motoring and less at heavy load than at light load, Kadenacy effect is responsible. If the Kadenacy effect is less pronounced and or the exhaust back pressure is considerable (small and or long exhaust pipe), the air-box pressure will begin to increase at an intermediate load. If the Kadenacy effect is very pronounced, the air-box pressure will be minimum when the load is maximum. These tests are described in more detail in Chapter 16.

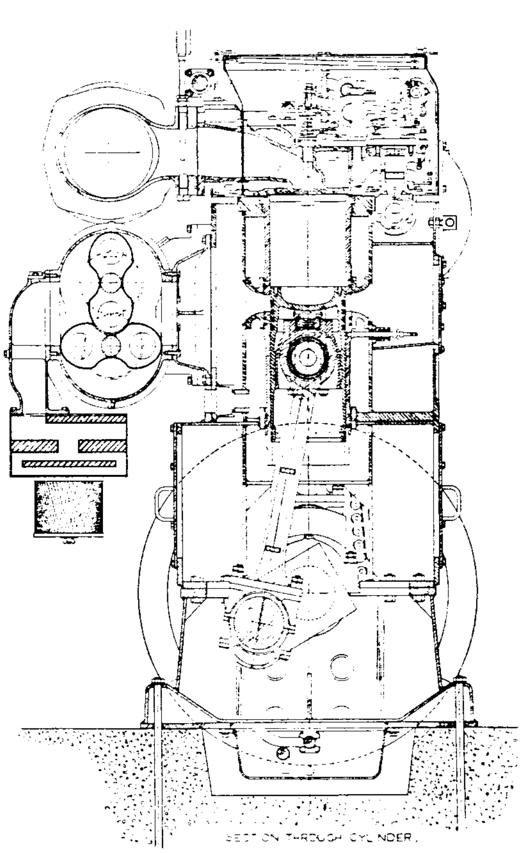

Fig. 13-4. Petter Superscavenge Engine. (By permission of Petters Ltd.)

CONSTRUCTIONAL FEATURES OF KADENACY ENGINES

13.6 Petter Engine.

While results obtained with the Kadenacy system are fully discussed in the literature [Dale, 1945], few of the constructional details have been disclosed. Figure 13–4 shows the longitudinal cross section of a modern Kadenacy type engine, the Petter Superscavenge Engine. It has $8\frac{1}{2}$-inch bore, 13-inch stroke, and normally operates at 600 rpm. It is built with 2, 3, 4, and 6 cylinders. It is rated at 85.88 psi bmep at which load the exhaust is said to be absolutely invisible. All engines carry the British Standard 10 per cent overload for one hour, and also a peak torque corresponding to 110 psi bmep or 30 per cent overload for considerable periods. The specific fuel consumption at rated load and speed is 0.367 pounds per bhp-hr with fuel of 19300 Btu heating value and the lubricating oil consumption less than $1\frac{1}{2}$ per cent of the fuel. The blower pressure used is only 2.8 psig at 600 rpm.

An examination of the section shown in Fig. 13–4 reveals a more or less conventional engine. Neither do the drawings and photographs of the components show very unusual features. Both the inlet and exhaust ducts are carefully streamlined. The exhaust duct is divergent up to the exhaust header. Roots blowers are mounted on the side of the engine, each serving two cylinders. The inlet ports are made with various angles as shown in Fig. 13–5 which is claimed to cause the entire cylinder charge to rotate without leaving a stationary core in the center.

The exhaust is released through two valves in the head of rather large diameter and high lift but otherwise conventional.

The piston shown in detail in Fig. 13-6 is oil-cooled, of aluminum alloy, with fixed wrist pin clamped in an aluminum insert bolted to the piston body which is without the usual holes. Oil is fed from the rod by a pipe from the crank pin bearing to the bushing of plastic bronze which is free to turn between the wrist pin and the carbon-chromium steel bushing fitted to the eye of the connect-

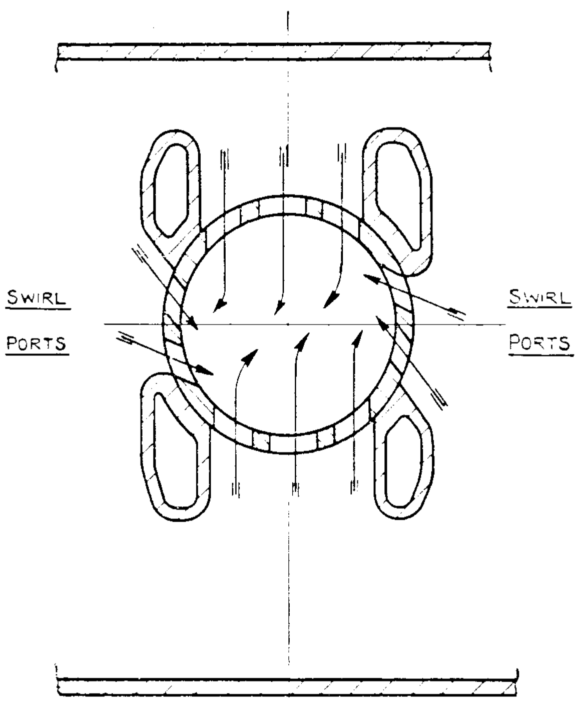

Fig. 13-5. Inlet Ports of Petter Superscavenge Engine. In this section through cylinder and liner arrows indicate the believed air flow causing rotation of charge.

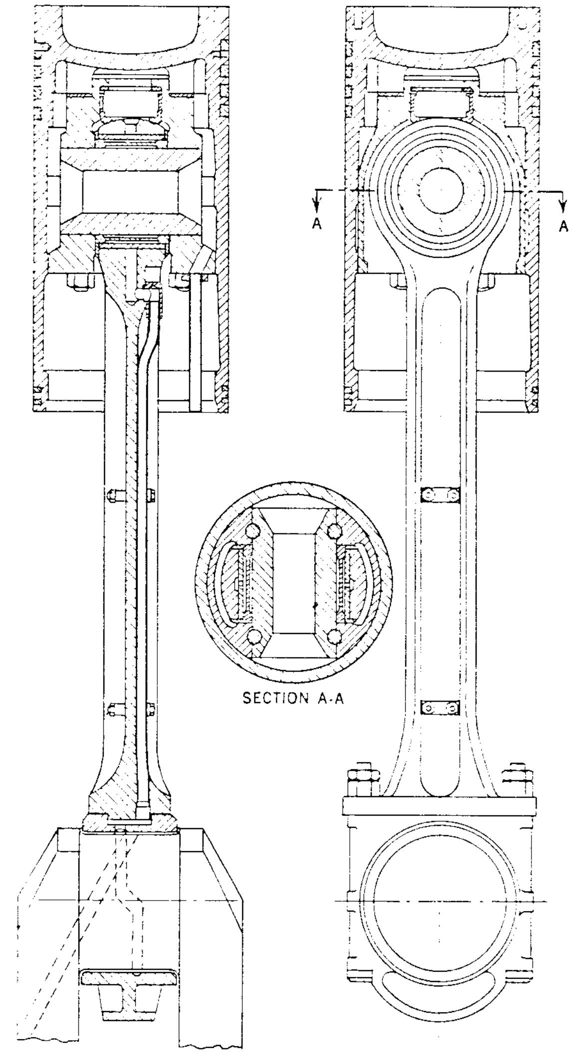

Fig. 13-6. Assembly of Piston and Connecting Rod Petter Superscavenge Engine.

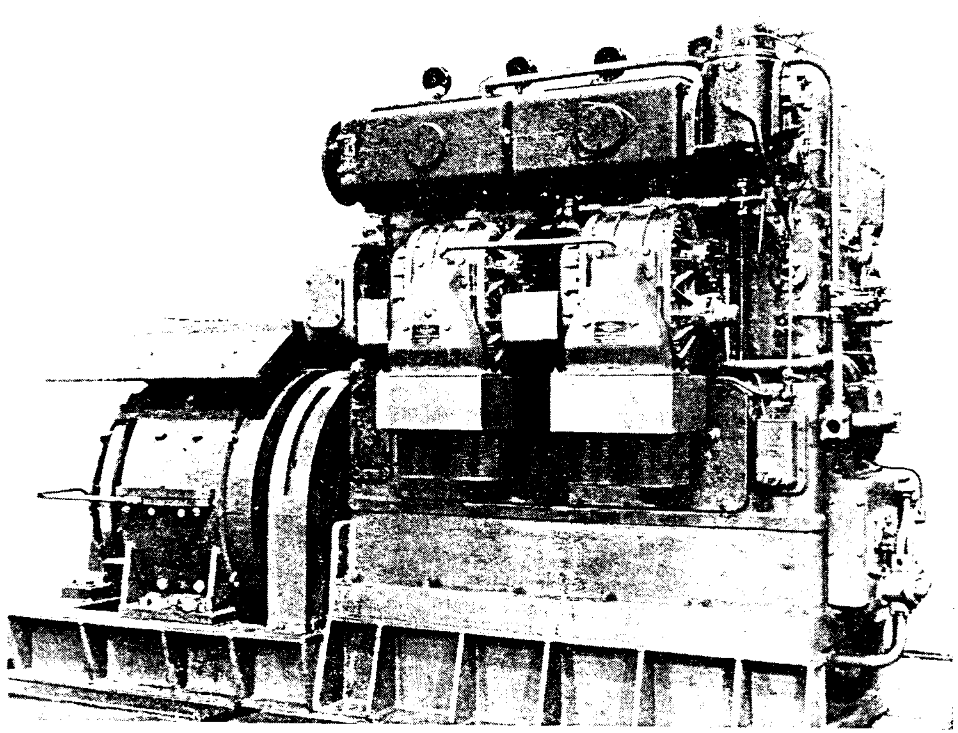

Fig. 13-7. Back View of a Four-Cylinder Petter Superscavenge Engine.

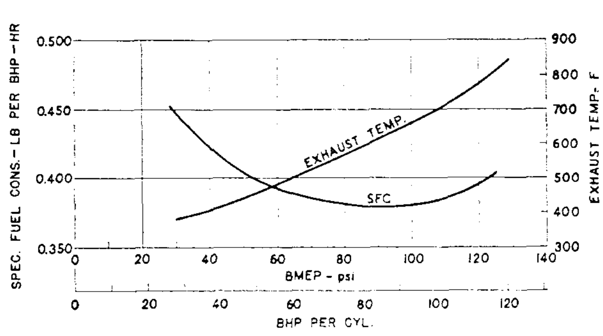

Fig. 13-8. Specific Fuel Consumption and Exhaust Temperatures of Petter Superscavenge Engine.

ing rod. The piston-cooling oil passes from the channels in the bushing and the rod to the crown through a sealing piston with a spherical seat at the top of the rod, to which it is spring-loaded.

A back view of a four-cylinder auxiliary generator set is shown in Fig. 13–7 and Fig. 13–8 shows the specific fuel consumption and exhaust temperatures of this engine plotted against load.

Figure 13–9 shows a weak-spring indicator diagram of the engine, showing the slight dip below the atmospheric line after the completion of the blowdown period, which is characteristic of the Kadenacy system. This dip is much more pronounced in the diagrams of blowerless types of Kadenacy engines some of which are shown by Davies.

The inlet duration is 100 degrees and the exhaust lead 21 degrees.

DESIGNING FOR KADENACY EFFECT

13.6a It has been fairly well established that the Kadenacy effect cannot be dismissed as fictitious as it is based upon a demonstrable phenomenon: the rarefaction that follows sudden discharge of compressed gas from a closed vessel. Both thermodynamic theory and engines built according to the Kadenacy system prove the soundness of this principle.

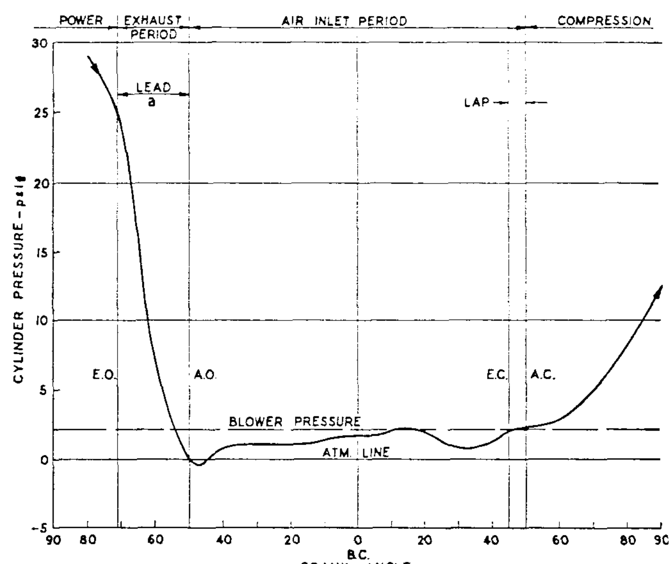

Fig. 13-9. Weak-Spring Indicator Diagram of Petter Superscavenge Engine. (B. M. Dale, "Recent Development in Medium-Speed Two-Cycle Oil Engines." *Motor Ship*, **25**, No. 300, Jan., 1945.)

The Kadenacy effect is not due to the inertia of the gas column in the exhaust pipe but to the inertia of the gas column in the cylinder, and arises therefore with any length of exhaust pipe if conditions are favorable. It is true, however, that gas inertia in the exhaust pipe can give powerful assistance to the Kadenacy effect if the pipe is properly tuned and destroy it if it is not.

All well-designed two-stroke cycle engines are benefited more or less by the Kadenacy effect, but in order to take full advantage of it, the following five points deserve particular consideration: (1) proper exhaust lead, (2) rapid exhaust opening, (3) streamlining of flow passages, (4) exhaust tuning, (5) elimination of reverse flow.

13.7 Proper Exhaust Lead.

The exhaust lead must be long enough to permit the cylinder gas to discharge from the cylinder and create a depression in its wake before much inlet air is admitted, but not so long that a return

wave from the exhaust pipe would reach the exhaust orifice while it is still open and readmit the burnt gases to the cylinder.

For the calculation of the exhaust lead Kadenacy gives the following formula [U. S. Patent No. 2,144,065]

$$\frac{W}{100K\,Av} = \frac{a}{360N} \tag{13-1}$$

where W is the cylinder volume in cubic centimeters, A the area of exhaust lead in square centimeters, v a *hypothetical* mean velocity of mass exit of the burnt gases of the order of 450 meters per second, K a constant depending upon the form of the exhaust orifice and the area opened per unit movement of the piston or crankshaft, a the angle of exhaust lead in degrees, and N the speed of the engine in revolutions per second. With the symbols used in this book

$$KA = A_m$$

$$a = \alpha$$

$$N = \frac{n}{60}$$

and

$$W = V_e$$

equation (13-1) can be written as

$$A_m\alpha = \frac{6n}{100}\,\frac{V_e}{v}\ \text{cm}^2\ \text{deg.} \tag{13-2}$$

For the *hypothetical mean velocity of mass exit* Kadenacy recommends 450 meters per second but also mentions that it was found in practice that it should be in the neighborhood of 500 meters per second. Taking $v = 475$ meters per second equation (13-2) becomes

$$A_m\alpha = \frac{V_e n}{7900}\ \text{sq cm deg}$$

or in English units

$$A_m\alpha = \frac{(2.54)^3}{(2.54)^2}\,\frac{V_e n}{7900} = 0.00032\ V_e n\ \text{sq in. deg.} \tag{13-3}$$

This is practically identical with equation (8-1) which has been derived in the appendix of Chapter 8, without any assumption of *ballistic* velocities.

The proper exhaust lead for Kadenacy effect can therefore be calculated by the method described in 8.2.

According to either equation the required blowdown time area is proportional to the engine speed: therefore the *minimum* exhaust lead required to insure evacuation of the cylinder increases with the speed.

The time required for the return wave to reach the cylinder is approximately constant in terms of seconds but in terms of crank degrees it also increases directly with the engine speed.

It is seen that both the minimum and the maximum exhaust lead increase with the engine speed. In a variable-speed engine optimum exhaust lead can be designed for only one given speed. If the engine speed much exceeds the design speed, the blowdown will be incomplete and the Kadenacy effect is largely lost. If the engine speed is much lower than the design speed, the return waves from the exhaust pipe find the exhaust orifice still open and fill the cylinder with exhaust gases before fresh air has an opportunity to get in.

Kadenacy's U. S. Patent No. 2,113,480, pictured in Fig. 13–10, shows a mechanical arrangement for varying the exhaust lead to suit the engine speed. No such engine is known to have been built.

It may be concluded that the Kadenacy effect can best be applied to a constant-speed engine. In a variable-speed engine the exhaust lead must be designed for the preferred engine speed, and the calculation then is identical with the one described in Chapter 8. It requires, however, skill in design to insure adequate exhaust lead in a high-speed engine without unduly crowding the inlet period.

13.8 Rapid Exhaust Opening.

In all but very slow-speed engines rapid opening of the exhaust is desirable in order to provide adequate blowdown area without unduly reducing the duration of the inlet. With the Kadenacy system rapid opening of the exhaust is not only desirable but essential because the Kadenacy effect vanishes if the discharge opening is not fast enough.

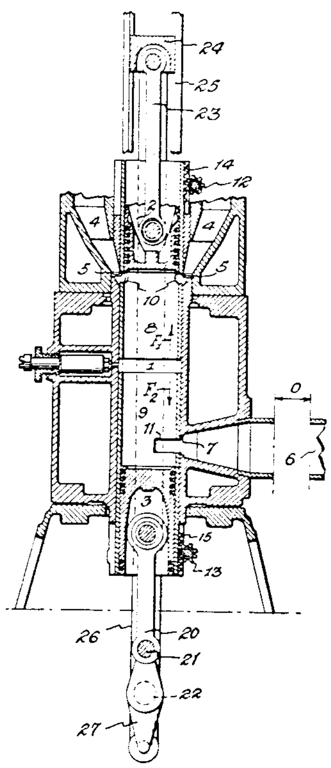

Fig. 13-10. U. S. Patent No. 2,113,480, M. Kadenacy. By pinions 12 and 13, and racks 14 and 15 the exhaust lead is so varied that the desired timing is obtained for varying engine speeds.

For the speed of opening the exhaust Kadenacy fails to give definite rules, but indicates (U. S. Patents No. 2,144,065 and 2,123,569) that the *critical exhaust lead* should be less than 1/300 second and preferably as low as 1/500 second. The critical exhaust area is given as approximately one-quarter of the piston area and should be uncovered in less time than that required for the blowdown period. The Penn State tests on the other hand have shown that the larger the suddenly opened exhaust orifice the greater is the ensuing depression. By reducing the exhaust opening from 100 per cent to 45 per cent piston area, the depression at 5 atmospheres chamber pressure dropped from 4.3 to 1.7 psig—that means to 40 per cent—and at 25 per cent piston area the depression was only 0.85 psig, or only 20 per cent of the maximum. It is obvious that a rapid opening is essential for the Kadenacy effect and the more rapid the opening the better the effect.

13.9 Mechanical Difficulties.

The mechanical difficulties involved in producing a very sudden exhaust opening are, however, considerable. Instantaneous opening of a large area in an engine cylinder cannot be realized or even closely approached by practical means. The uncovering of as much as one-half of the piston area is out of the question and even the opening of a smaller area involves appreciable time.

To satisfy Kadenacy's rather limited objective, one-quarter of the piston area must be uncovered during the exhaust lead. Piston-controlled ports are considered first. In an opposed-piston engine the rectangular exhaust ports may occupy as a maximum 75 per cent of the circumference. If the height of that portion of the exhaust ports which is uncovered before the inlet ports open is designated h_b,

$$0.75D\pi h_b = 0.25D^2\frac{\pi}{4}.$$

Assuming a stroke length of

$$s = 1.2D$$

$$D = \frac{s}{1.2}$$

$$h_b = \frac{D}{12} = \frac{s}{12 \times 1.2} = \frac{s}{14.4} = 0.07s.$$

An inlet duration of 100 degrees crank angle is assumed. From Fig. 7-5 this corresponds to 14 per cent stroke port height. Adding to this for h_b, 7 per cent stroke, 21 per cent stroke is shown. This would make the exhaust ports open at 119 degrees after top center, which is acceptable.

The exhaust lead consisting of $180 - \frac{100}{2} - 119 = 11$ degrees must be covered in less than, approximately, 1/300 second. From the relation

$$11 = 6n \times \frac{1}{300}$$

the engine speed must be higher than 550 rpm.

Considering next a cross- or loop-scavenged engine, the exhaust ports cannot occupy more than 30 per cent of the circumference and h_b must satisfy the following requirement:

$$0.3D\pi h_b = 0.25D^2\frac{\pi}{4}$$

which, with $s = 1.2D$,

$$h_b = \frac{s}{(0.3)\times(16)\times(1.2)} = 0.173s$$

or 17.3 per cent of the stroke. With 100 degrees inlet duration the upper edge of the exhaust port would be at 31.3 per cent from the bottom of the stroke, which makes the exhaust ports open at 105 degrees after top center, which is somewhat too early for a port-scavenged engine, but still acceptable.

The exhaust lead now is 130 − 105 = 25 degrees, and this must be covered in not more than 1/300 second, which means that the engine speed must be higher than 1250 rpm. This would indicate that the Kadenacy system is not suited for cross- or loop-scavenged engines.

The final example is a cylinder with 2 poppet exhaust valves, the diameters of which are 40 per cent of the piston diameter. In order to provide a valve opening equal to one-quarter of the piston area

$$2 \times 0.4\,D\pi \times l_c = \frac{1}{4}\frac{D^2\pi}{4}$$

from which the *critical* valve lift is

$$l_c = \frac{D}{16}\frac{1}{0.8} = 0.078D$$

Although this is a fairly high lift, it is not the total lift required, but only that portion of it which corresponds to the exhaust lead period. In order to provide the necessary deceleration, the entire valve lift may have to be more than this.

The critical valve lift must be completed in less than 1/300 second. With a 6-inch bore cylinder the valves must travel 0.48 inch in 0.0025 second. Starting from standstill and assuming uniform acceleration

$$0.48 = \tfrac{1}{2}a(0.0025)^2$$

from which

$$a = 154{,}000 \text{ in./sec}^2 = 396g$$

which is very high indeed.

In ordinary valve gears, accelerations do not reach more than a fraction of this figure and even in engines built after the Kadenacy principle the valve accelerations and consequently the critical valve lifts are considerably below the minimum required by Kadenacy's patents. Instead of 1/500 to 1/300 second, the periods of critical exhaust opening vary between 1/100 and 1/200 second and even these figures require extraordinary valve gear design.

13.10 Masked Valve.

Various expedients are being used to avoid the enormous valve accelerations, and correspondingly enormous spring loads. One of them is the use of *masked* valves.

If the velocity of the valve is zero at the beginning, the opening is very small for the first 20 degrees. By recesssing the valve seat in such a manner that the outer diameter of the valve constitutes an easy fit in the recess, with a diametral clearance of approximately 0.030 inch, the valve acts as a piston valve and it is possible to start the motion earlier. Such a valve is shown in Fig. 13-11.

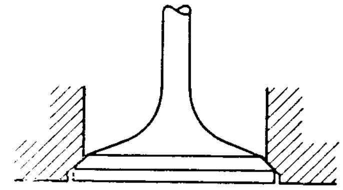

Fig. 13-11. Masked Valve.

The effective exhaust begins only when the valve head clears the recess. In this way the effective opening of the exhaust may remain the same but the time available is increased. In this way the acceleration can be greatly reduced.

A disadvantage of the masked valve is that the total valve lift is still greater. Another disadvantage is that the hot gases rushing through the small annular area between the cylindrical part of the valve and the recess are pretty hard on the seat.

Other expedients, such as piston valves and combination valves, have also been proposed for keeping the accelerations within bounds. If high accelerations are unavoidable, recourse is taken to

unconventional springs such as multiple-coil, hairpin, and torsion-rod types to provide the very high spring load required for handling the high accelerations.

It is evident from the foregoing that the task of the designer to provide exhaust openings large enough and fast enough for the effective utilization of the Kadenacy effect is a difficult one. It may even be said that this difficulty constitutes the greatest obstacle to full exploitation of the Kadenacy effect. However, even a partial utilization of the Kadenacy effect is worth while, especially with blower-scavenged engines. Therefore, the above-cited figures should not be interpreted as a deterrent to any effort directed to the utilization of the Kadenacy effect.

13.11 Streamlining of Flow Passages.

It is natural that when the flow of air is controlled by low pressure differences, as is the case with the Kadenacy system, conservation of the kinetic energy in the stream is very important. Both the exhaust and inlet passages must be *streamlined* by providing ample cross-sectional area, avoiding sharp turns, protrusions, and other eddy-producing shapes. For inlet ducts and ports slightly convergent shapes are chosen to provide guidance for the air in the cylinder. For exhaust ducts slightly divergent shapes give good results because of the aspirator effect of the divergent tubes. Figure 13–4 shows an example of good streamlined duct forms.

For exhaust, tapered pipes have also been used, with an included angle of a little over 1 degree.

13.12 Exhaust Tuning.

Successful application of the Kadenacy system is greatly promoted by tuning the exhaust pipe. This is particularly true of the blowerless type of Kadenacy engine, where exhaust tuning can be of decisive importance. In multicylinder engines, with more than three cylinders, divided exhaust lines are generally used to avoid interference among the cylinders and to control better the pressure pulsations in the exhaust ducts.

For proper tuning the analytical methods described in Chapter 12 may be used and should be supplemented by the experimental methods described in Chapter 16. Tuning of a Kadenacy type engine does not differ from that of a conventional engine, except that with the former, exhaust tuning is more essential than with engines that depend on relatively high scavenge pressures for charging.

The necessity of proper exhaust tuning again tends to make the Kadenacy engine a single-speed machine. Naturally the design speed for port timing must agree with that of the piping. Operation at other speeds may be possible but less good performance must be reckoned with.

13.13 Elimination of Reverse Flow.

It has been pointed out that the Kadenacy effect can easily be destroyed by a reverse flow generated by a reflected pressure wave from the end of the exhaust pipe or from a receiver in the exhaust line. In such a case the charging of the cylinder with fresh air is spoiled by the return of burnt gases which mix with the fresh air and/or prevent its entry into the cylinder.

This is likely to happen if the exhaust pipe is so short that the return wave finds the exhaust orifice still open. To eliminate this contingency the relation of the natural frequency of the pressure wave and the engine speed must be such that the period of the exhaust pressure wave shall be longer than the duration of the exhaust opening. Methods for accomplishing this are described in Chapter 12.

Neither is it desirable to have the depression created by the exhaust column persist so long that it sucks out of the cylinder whatever fresh air has been admitted. Such effect would result from too long an exhaust pipe.

Good results are obtained if the exhaust pipe is so tuned that the period of one complete oscillation approximately equals the duration of the exhaust, as was explained. But this is possible only for one speed. The question is how to make the Kadenacy system operable at variable speeds without contending with the obstruction or contamination caused by return of the burnt gases at certain engine speeds.

Solutions have been offered toward converting the exhaust duct into a one-way thoroughfare. The outwardly tapered exhaust pipe mentioned above may be considered such a device inasmuch as it discourages flow in the converging direction.

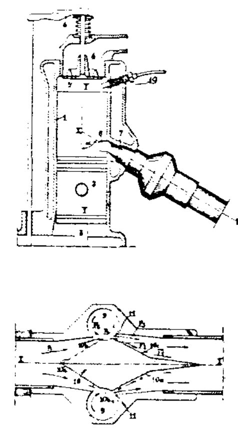

Fig. 13-12. U. S. Patent No. 2,110,986. M. Kadenacy. Outward flow at the exhaust gases obstructed, flow stopped by a "plug" of whirling gases in toroidal cavity 9.

A more positive action could be expected from a nonreturn valve which would allow free passage to the outward flow of the exhaust gases but would prevent by its automatic closure any return flow. The objections to such valves so far proposed are of mechanical nature. The speed of operation, the high temperature of the exhaust gases and the smoke and soot which they contain make very difficult conditions for a valve to operate under. Therefore exhaust devices involving moving parts met with little success so far.

Kadenacy, in his numerous patents, proposes various stationary devices to accomplish such purpose. Figure 13–12 shows a deflector device to be interposed between the exhaust duct and manifold, that is supposed to offer little resistance to the outward flow of the exhaust gas. The return flow, however, caused by the rebound of the gas column from the end of the pipe, encounters a gaseous *plug* formed by the whirling gases in the toroidal cavity 9, located as close to the cylinder as possible.

Somewhat similar devices are shown in Fig. 13–13 and 13–14.

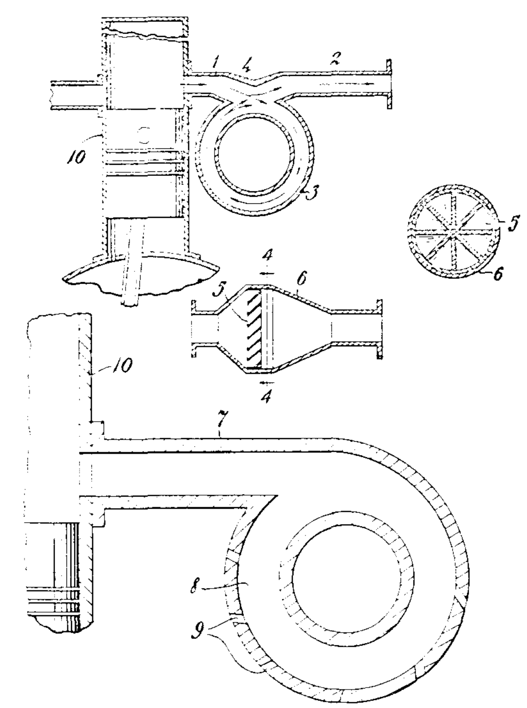

Fig. 13-13. U. S. Patent No. 2,167,303, M. Kadenacy. Means to minimize the rebound of the exhaust gas column by prolonging its outward motion and delaying its reversal and or reducing its intensity.

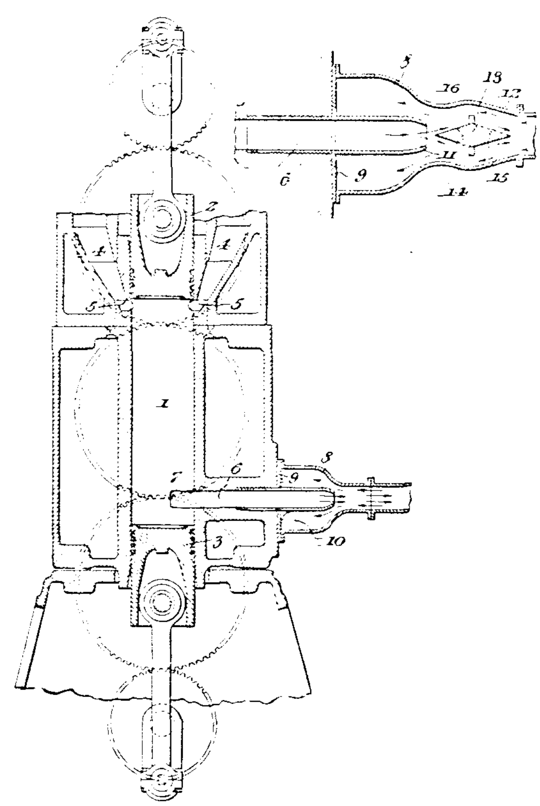

Fig. 13-14. U. S. Patent No. 2,168,528, M. Kadenacy. Means to prevent a too early return of the exhaust gas column that would reach the cylinder when the charging process is still in progress.

Reports of the successes of the above-described devices are not sufficiently conclusive to have them recorded.

Two-Stroke Power Units, P. E. Irving
London: Temple Press Books, 1967

The Kadenacy Effect

It was long thought and even openly stated that the two-stroke would always be limited to low speed and hence low power, just because of the limited time during which the ports were open, but this view is fallacious and in fact the opposite is true; it is *because* the ports – or rather the exhaust port – opens so rapidly that very high power can be developed even at speeds exceeding 10,000 r.p.m. when this port is open for less than 0·003 seconds, of which time only about half is utilized for discharging the cylinder.

This is because of a phenomenon known, after the worker who first investigated it, as the 'Kadenacy effect'; he demonstrated that when an exhaust port is opened extremely rapidly, the imprisoned gas bursts out with such velocity that it in effect 'overshoots' and the pressure in the cylinder is reduced below that of the atmosphere; the quicker the rate of opening and the higher the initial pressure the more intense does the effect become, and a negative pressure of up to 6 p.s.i. may be created.

It is really this self-emptying process rather than the entry of fresh gas from the transfer ports, which is responsible for the scavenging of the cylinder, especially at part throttle when there is insufficient fresh charge coming in to displace much of the spent products. The purpose of the transfer-port or deflector shaping is mainly to ensure that there is as little mixing as possible between the two lots, and as Dr. Ehrlich puts it, 'to direct an unbroken column of fresh charge straight up towards the head'.

The reduction of pressure following the initial discharge will cause the cylinder to fill up again from somewhere; normally this will be via the transfer ports but it could also be by back-flow through the exhaust port. Indeed, if the exhaust pipe is very short or non-existent and open to the atmosphere, fresh air may enter and dilute the mixture so much that it will not subsequently fire. As described in Chapter Six the Kadenacy effect can be augmented by the characteristics of the exhaust system and furthermore advantage can then be taken of the lag between the transfer and exhaust closing points to provide very high power. The converse is also true; an incorrect system can utterly ruin the performance of any engine, irrespective of its design.

The Crankcase Inlet Port

A piston-controlled inlet port is timed to be opened and closed by the skirt at somewhere between 60° and 75° from the top dead centre position, the actual figure again being determined by the use for which the engine is intended. During the first part of the outstroke of the piston the transfer port is still open and therefore the piston cannot commence to generate a depression in the crankcase until this port is

closed or at least nearly so; following that, in the 60° or so of rotation until the inlet port commences to open, a depression of 5 or 6 lb. below atmospheric pressure will have been created and fresh mixture rushes in from the carburetter. At low speeds and open throttle, the crankcase will be virtually filled by the time t.d.c. is reached; after that the descending piston commences to compress the crankcase charge, some of which will tend to blow back through the still-open inlet port and the carburetter, causing an undesirable loss of fuel. This is an inherent bad feature of symmetrical inlet timing, but its effect is less at part-throttle because the crankcase pressure does not build up so rapidly, and also is less at speed, because in order to blow back, the crankcase pressure has to reverse the direction of flow of the incoming gas which is moving inwards at a velocity of several hundred feet per second. By using a long induction pipe, so that the moving column of gas possesses a greater amount of inertia, the 'ram' effect created will improve crankcase filling markedly at high speeds (see Chapter Nine).

It is clear, however, that the inlet port timing must be determined in relation to the speed at which the engine is required to give its optimum performance; at any other speed it is something of a compromise. One common way of reducing blow-back while still retaining a longish period of opening is to make the lower port-edge of a flat V or curved shape or notched so that as the area approaching closing point is reduced the incoming charge-velocity remains high, even though it has slowed down in the actual inlet pipe. This scheme also reduces the intensity of the 'plopping' sound set up when the port opens: the noise arising from the rapid opening of a square-edged port can be very objectionable and moreover is difficult to silence without causing an undesirable restriction to gas-flow. In the B.S.A. 'Bantam', the rectangular portion of the V-notched port is fully closed at 50°, closure of the notched portion occupying a further 10° of crank travel.

Non-symmetrical Timing

In order to circumvent the ills which were laid at the door of symmetrical port timing, various methods have been adopted based on the use of twin-piston cylinders, while many varieties of automatic or mechanical valves have been, and some still are, used instead of a plain inlet port to the crankcase. One method requiring no additional complication is to make the cylinder axis offset or '*desaxé*' in relation to the crankshaft; if the offset is forwards and about 15 per cent of the stroke (the usual figure) the exhaust and transfer ports will open later by about 4° (depending on the con rod/crank ratio) and close later by roughly the same amount, which seems to be the wrong way, but the inlet port will open later and close earlier which is correct. To reverse these effects, the cylinder would need to be offset to the rear, thereby possibly improving the power port timing but not the inlet port timing,

CHAPTER TWO

PORTS AND VALVES IN DETAIL

SINCE the time available for emptying and filling a two-stroke cylinder is so much less than in a four-stroke (in which, moreover, the two processes do not occur simultaneously, but in succession), it follows that the timing of the ports is much more sensitive than the timing of valves; a difference of say 10° in the exhaust-valve opening-point would make very little difference to the performance of a four-stroke whereas it could utterly ruin that of a two-stroke, especially if the transfer-port timing was not altered to suit. In the early days, before precision-casting methods had been developed, unavoidable errors in port-location and shape led to great discrepancies between the performance of engines which were supposed to be identical. Foundry technique has greatly improved since then, but even today many makers prefer to incur the additional expense of machining the ports, either by drilling multiple holes through the walls, or by forming shaped ports in a liner before this is fitted.

The Exhaust Port

In its simplest and also most efficient form, this is a single orifice, of roughly rectangular or elliptical shape, blending into a circular pipe connection; the height above the piston crown edge at b.d.c. is determined by the timing required and the lower side should not project above the crown at this point, though it does not matter if it is fractionally lower. The width depends upon the kind of performance required as a large port, while giving better flow area for high speed, is also likely to permit more charge to be lost at low speed. The maximum usable width is limited, by the tendency of the piston rings to spring outwards, to about 20 per cent of the circumference, and if a greater width is necessary, it must be split up, either by drilling two or more holes, or by casting a bar down the centre. If the cylinder is linered the bar should be backed up by a streamlined division cast into the port.

Ports with curved or sloping edges (Fig. 2.1) deal more kindly with the rings than those in which the edge is straight across, because these fragile components are eased back into their grooves more gently and there is no chance of them striking against the whole width of the port simultaneously. The rate of opening or rather the rate at which the effective area increases during the initial few degrees of the opening

period is also more gradual; this slows down the rate of discharge of the gas during the blow-down period and makes the exhaust note easier to silence, but also reduces the Kadenacy effect and the pressure intensity of the exhaust pulse which is used to produce high power, especially in racing engines (Chapter Twelve). Racing engines therefore usually have rectangular ports and it is usual, when modifying 'cooking' engines for speed, to square-out circular or 'gabled' ports, thereby increasing the time-area integral without altering the timing. As far as the rings are concerned, it helps to relieve the bore along the upper edge slightly, but overdoing it will adversely affect the low-speed running.

Deposits of carbon or partially burnt oil form on the port walls in time and may seriously curtail power output; the greater the surface area in relation to the cross-section the worse this effect will be and the single plain port is less liable to blocking than one with several small holes, though fortunately the trend towards low oil/fuel ratios and improved lubricants has greatly reduced this once-prevalent trouble. Carbon which builds up into a long, rigid deposit is difficult to dislodge, but if the ports are formed in a thin liner, with the outer sides bevelled

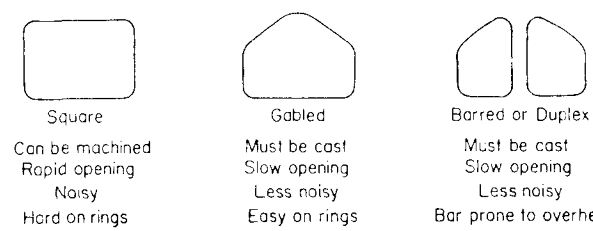

FIG. 2.1. Features of three common shapes of exhaust ports.

and opening into a large space, any thin skin of carbon which may build up on the edges during light running will be broken off after a few revolutions at full throttle, so the ports are virtually self-cleaning – a valuable asset in marine or stationary engines which may be required to run non-stop for long periods without loss of performance.

To reduce the absorption of waste heat, the external port should be as short as possible, especially in air-cooled engines; it is better to cut the cylinder fins away locally to attain this end, rather than to use a long port to which the fins are attached. On motorcycles, it is usual to place the port at the front, where it appears to receive the most cooling air, but on racing machines this is not always so, due to shielding by the front wheel. On the air-cooled M.Z. and Suzuki machines, with heavily finned aluminium barrels, the ports are at the rear, which simplifies the exhaust system and was also found to obviate the piston-seizures which occurred with the ports in front.

CHAPTER SIX

EXHAUST SYSTEMS AND SILENCING

IT has always been recognized that a two-stroke is particularly sensitive to its exhaust system; the [redacted] in the woodpile was thought to be 'back-pressure' caused by resistance to gas-flow. Very free systems were often used, which always created a great deal of noise but did not necessarily improve performance and, in fact, often diminished the power and increased the fuel consumption at the same time.

In Chapter One it was explained that even without an exhaust pipe, the Kadenacy effect created by a rapidly opening port is itself capable of reducing the cylinder pressure to well below atmospheric. This effect can be either assisted or its results nullified by the characteristics of the whole system from port to air, which may include the primary portion of pipe, one or more expansion chambers with or without baffles, and a tail pipe or at least a fish-tail or similar restriction. Taking the simplest case, merely a parallel, open-ended pipe, the general sequence of events is this. The puff or slug of high-pressure gas liberated through the exhaust port travels down the pipe at high velocity, the front of it forming a pressure-wave which travels at the speed of sound until it reaches the open end of the pipe. Meanwhile, the tendency for the slug to keep moving has created a depression behind it, thereby increasing the intensity of the Kadenacy effect and reducing the cylinder pressure to several pounds below atmospheric. There is thus a positive pressure-wave moving outwardly and a negative wave moving in towards the cylinder, the latter assisting the crankcase pressure to bring in fresh mixture through the transfer ports. The pressure wave, on reaching the open end of the pipe, obeys a fundamental law of acoustics and is reflected back as a *negative* wave, of reduced intensity because much of the wave energy is lost in the surrounding atmosphere. However, some energy remains and the wave travels back to the exhaust port. If this is still open, gas which will be partly burnt and partly fresh mixture (depending on circumstances and the effectiveness of the cylinder scavenging arrangements) will be drawn out into the pipe; meanwhile, the negative wave is reflected from the cylinder, still *negative* because the cylinder forms a 'closed' end to the system, travels back to the open end, is reflected as a *positive* wave and returns to the port, all this occurring at the speed of sound, which is around 1,400 ft. per sec. in hot exhaust gas but increases with temperature and diminishes with pressure. If at the time of arrival the port is fully closed, the wave just

bounces back again, taking with it any of the fresh charge which had been drawn out previously and which will be irretrievably carried away by the next exhaust pulse, but if the port is *not* fully closed there will be a back-flow into the cylinder of gas which will contain at least a portion of the escaped charge which will thus be used to produce power on the next stroke and a loss will have been turned into a profit. It will be seen that for the return pulse to be effective the wave has to traverse the pipe *four times* in the interval between the port being opened sufficiently for the first pulse to build up in the pipe and it being not quite fully closed. To take a simple example, imagine an engine with the exhaust port opening and closing at 75° and the transfers at 63°; the pulse will be well established at 70°, and its second return should arrive by the time the transfers shut, giving a total period of 135°. At 5,000 r.p.m., this is equal to $60/5{,}000 \times 135/360 = 0{\cdot}0045$ sec. In this period a wave travelling at 1,400 ft. per sec. will cover 6·3 ft. so the length of the pipe plus port should be $6{\cdot}3/4 = 1{\cdot}57$ ft. or 18·8 in. To this must be added half the diameter of the pipe to allow for 'end effect' making the final length 19½ in. in round figures; subtracting 1½ in. for the port itself, the pipe length then becomes 18 in. That this calculation is somewhere near the mark is borne out by the fact that the T.T. model Scott performed best with near that length and was given a special dispensation to permit it to be raced in that form instead of with the long pipes demanded by the race regulations then in force. Looking at the matter a little further, after the exhaust port closes at 75° any of the pressure-wave remaining will be reflected back and forth as before, altering in sign or becoming negative at the open end but not changing sign at the port which will remain closed for a further 210°. The wave will traverse the pipe six times and will be negative at the port end in $135° \times 6/4 = 202°$, or just about the time the port is due to open again, and will thus be able to assist the exit of the next exhaust pulse; in fact the whole system is then operating 'in resonance' and good power without excessive fuel consumption will be attainable.

This elementary open system has two defects; it is unbearably noisy, the note containing many high-pitched overtones of a particularly irritating quality, which puts it out of court for anything except racing machines, and also it is effective only over a limited range of speed, although not quite so limited as one might imagine (a) because the time of arrival of the exhaust port waves is not critical to a few degrees and (b) because at high speed and load, the exhaust gas temperature and thus the velocity of sound becomes greater and the speed can rise proportionately without loss of resonance. However, at speeds much above or below the resonant range the returning waves will either interfere with scavenging, giving reduced power, or may suck out much of the charge without subsequently returning it to the cylinder, giving reduced power and excessive consumption as well. At low speeds a

short open pipe of around 10 in. long, suitable for 8 to 9,000 r.p.m. may even have the effect of feeding some fresh air back into the cylinder, upsetting the carburation very badly and causing difficulty in getting the engine to take up the load without excessive clutch-slipping.

Even for racing the plain pipe has another defect: much of the wave energy is lost in the form of kinetic energy when the first pulse leaves the end and if the gas velocity is approaching the sonic velocity much of the outgoing wave will be swept out of the pipe. Friction against the pipe walls attenuates the wave-pressure and between the two things the eventual wave which reaches the port is considerably reduced in its intensity. Fitting a 'megaphone' or divergent conical end, or even tapering the pipe right from the port, decreases the amount of energy lost to the atmosphere, partly because the gas velocity at the exit and

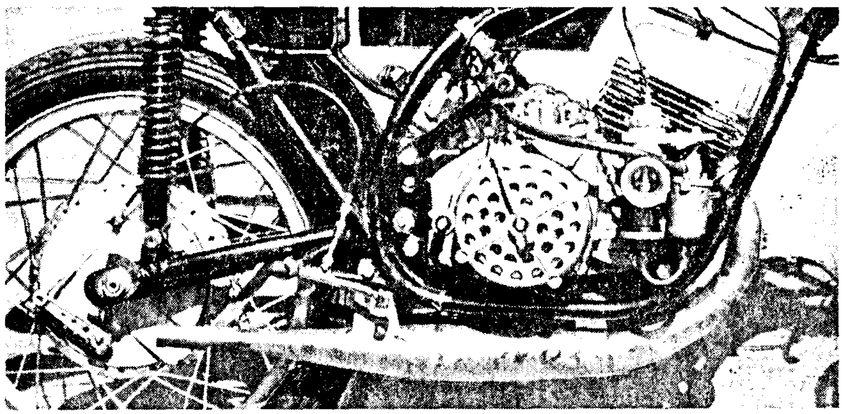

FIG. 6.1. One of the pair of expansion chambers on the 1963 250 c.c. twin Yamaha.

therefore its kinetic energy is very much reduced, and partly because the wave reflection does not occur precisely at the open end but takes place by stages along the length of the cone or 'megaphone'. In fact the process of wave formation and reflection is exceedingly complex, any increase in section (i.e. the junction of a pipe with a larger pipe or expansion chamber) causing partial reflection with change of sign, while a decrease in section acts to some extent like a closed end and causes partial reflection without change of sign. So will a drop in temperature, owing to the consequent drop in sonic velocity, and the mathematical derivation of even the natural or fundamental frequency of any system embodying a number of various diameters and volumes is complicated in the extreme. Much research has been done on the subject (*see* Bibliography) but unfortunately it is mostly presented in a form which is not immediately, if ever, clear to the practical engineer. Besides the natural frequencies, harmonics occurring at 3, 5, 7, etc. times the speed

are also present and the 'left-overs' or fragmentary waves still remaining from previous cycles also exercise some effect, even if small.

The Expansion-chamber System

Undoubtedly, the most spectacular exploitation of exhaust wave phenomena is to be found in the expansion-chamber system used on modern road-racing motorcycles (Fig. 6.1). Externally this resembles an ordinary touring silencer with a tail pipe rather smaller than usual; internally there are no baffles and there is only a few inches of parallel pipe before the first taper of the expansion chamber commences. The volume of the chamber is considerable, being about eight to twelve times the cylinder volume and its capacity and proportions vary from make to make. Preferably it is of circular section, though it is sometimes made elliptical in order to fit into the space available; vibration set up by the intense wave-pressure has however often led to this shape splitting at the seams, which causes an immediate drop in power. The operation of this device is illustrated diagrammatically (with the expansion chamber shown at much reduced scale in relation to the engine) in Fig. 6.2 which has incidentally popped up in all sorts of publications but was originally published in *Motor Cycling* to illustrate an article written by the author. The shock wave created by the exhaust pulse at high rates of port-opening enters the system (Fig. 6.2a) and travels down it at the speed of sound leaving behind it a negative wave which reduces the cylinder pressure to below atmospheric through the Kadenacy effect already described. Meanwhile the transfer ports have commenced to discharge fresh mixture into the cylinder (Fig. 6.2b) transfer being assisted by the increased pressure-difference between the crankcase and the area adjacent to the exhaust port from about 6 p.s.i. to possibly 12 p.s.i. A part of this charge is actually drawn out through the port into the cone of the expansion chamber, but while this action has been going on the pressure-wave has been approaching the end of the chamber, where it builds up a greater pressure due to the restricted area of the outlet, which acts in effect as a 'closed' end. The wave therefore bounces back with considerable vigour, as a powerful *positive* wave, and not as a comparatively weak negative wave as it would be if the pipe were open-ended. The return of this positive wave then pushes all the extracted charge back through the exhaust port into the cylinder (Fig. 6.2c); it will be noted that this action occurs when the wave has only traversed the system twice (once down and once back) undergoing only one reflection, instead of traversing it four times with three reflections as it does with a tuned open pipe. It therefore loses little energy in the process and, other things being equal, the required length of the system will be somewhere around twice that of an open pipe and in practice is in the region of 24 to 30 in. long. The tail pipe also exerts an effect by virtue of its own natural frequency, for if a pressure-wave

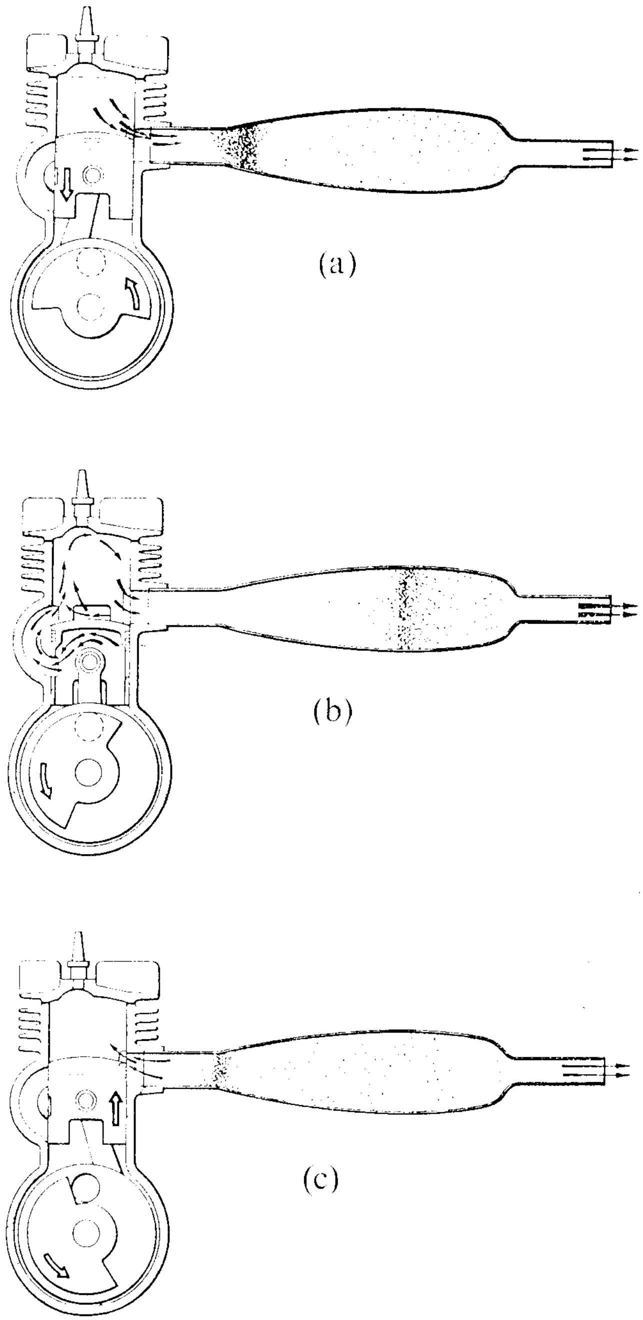

FIG. 6.2. The action of pressure-waves in the expansion chamber exhaust system.

travelling inwards through it arrives at the expansion chamber in time to meet the main wave on its way down, the latter will rebound with greater energy. Tail pipes have been known to split or break off short during races and this also causes drop in power output.

It is also necessary to maintain gas-tightness at the junction of the exhaust system, but any attempt to bolt the parts rigidly together always results in fatigue cracks due to the destructive high-frequency vibration in this region. One solution lies in using a double concentric spigot joint, with the exhaust pipe pushed over a stub and also fitting closely inside a sleeve, both stub and sleeve being welded to a flange: this construction permits angular or axial movement with little or no escape of gas.

Obviously this rather facile explanation is an over-simplification of the whole theory, which involves the natural and harmonic frequencies

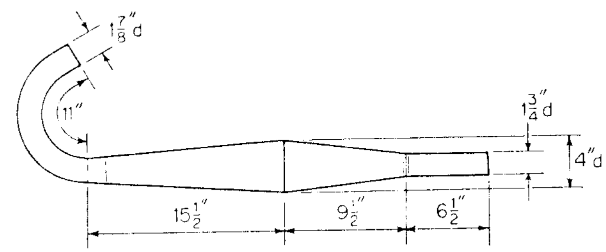

FIG. 6.3. Dimensions of an exhaust system suitable for the 250 c.c. Villiers competition engine.

of all the components involved. Walter Kaarden, who initiated much of the work on this system at M.Z., states that in the early stages, 'theoretical calculation of the dimensions did not agree with practical results, thus necessitating extensive tests with various designs to achieve optimum results' which at the time meant the attainment of a b.m.e.p. of 8 kg./sq. cm. or over 112 b.m.e.p.

Today powers approaching 220 b.h.p. per litre are being obtained from engines with cylinders of 50 to 125 c.c. with the aid of this system (plus of course correct design elsewhere) a figure which, as yet, cannot be attained by any four-stroke. These powers are developed in the 10,000 to 12,000 r.p.m. range depending on the cylinder size, but the usable power band is rarely more than 1,000 r.p.m. and often much less: below the critical, or resonant speed, the power almost ceases and the engine 'goes off the boil' unless the clutch is disengaged or a change made to a lower gear.

Ideally, the rising piston should close the exhaust port just as the last of the escaped charge has been forced back. If the port is late in closing—i.e., at lower speeds – the charge will be mixed with burnt

products. If the port closes too soon – i.e., at higher speeds – all the escaped charge will not be returned and some will be lost to the atmosphere; in either event, the power drops off sharply.

Expansion chambers ('resonance chambers' would really be a more suitable term) can however be used to good effect on less potentially powerful engines with 'softer' port timing and are even used on scramble engines where some degree of flexibility is required. Their bulk renders it difficult to install several of them within the small outlines of a modern racing-car although they have been fitted to three-cylinder Auto Union engines in this application. The introduction of the 1,000 c.c. capacity-limit for Formula 2 cars in 1964 opened up the possibility of a four- or six-cylinder engine, which with ported inlets and expansion chamber exhausts, should be able to develop 140 b.h.p.

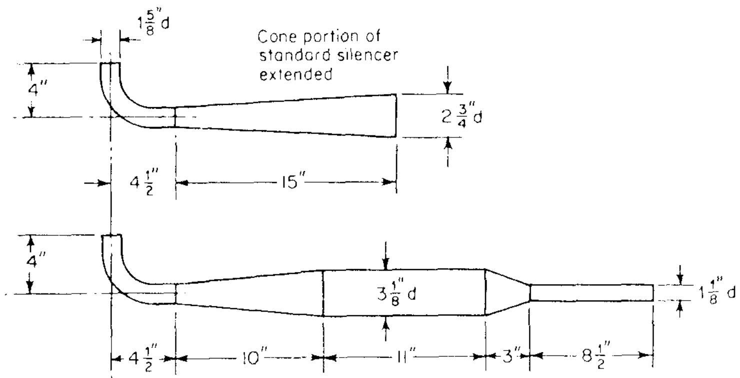

FIG. 6.4. Alternative racing exhaust systems for the Ariel 'Arrow', 250 c.c. twin. The megaphone (above) gives less power (23 h.p. at 7,400 r.p.m.) but greater flexibility than the expansion chamber (34 h.p. at 8,500 r.p.m.).

Exhaust Pulse Charging

When three cylinders exhaust into a common manifold advantage can be taken of the pulse of pressure from one cylinder to create a high pressure in the exhaust port of the cylinder immediately preceding the first one in the firing order, at or before the time at which that port is about to close. This will slow down or even reverse the direction of flow in the closing port with a reduction of charge loss and gain in power. This principle, referred to as 'exhaust pulse charging' has been used for many years in heavy Crossley marine engines, supplied with scavenging air at about 6 p.s.i., some of which is deliberately permitted to escape through the exhaust port and is then forced back again by the pressure-pulse from the next cylinder, firing 120° later. It is also used in the 3-cylinder Saab car engine, in which the exhaust port duration is approximately 160°, so that there is an overlap period of 40° during which two

out of the three ports are partially open simultaneously. By joining all three ports to a three-branch manifold, with all branches of equal length, the time taken for a pressure pulse to travel down one branch and up another can be adjusted so that the pulse arrives at the closing port at the appropriate moment, and by varying the length, which is only a few inches, the speed at which maximum torque is obtained can also be varied: above or below the selected speed, the effect becomes progressively less beneficial.

In the case of the Saab, an increase in b.m.e.p. from 70 to 86 p.s.i. at 3,000 r.p.m. is claimed by the use of this system, this increase being accompanied by a drop in specific fuel consumption from 0·72 to 0·65 lbs./h.p./hr. While not giving such a high power increase as that which would be gained with three tuned, separate pipes, the exhaust pulse system provides a less 'peaky' torque and is far less bulky and on both grounds is the more suitable of the two for use in a road vehicle.

Commercial Silencing Systems

For most situations other than racing, silence, or at least, comparative silence, is either obligatory by law or desirable from the operator's standpoint, and some method of muffling the sound without sacrificing too much power becomes essential.

In any case it is usual for the exhaust system to be limited in size or shape by some consideration other than maximum power and furthermore, for many applications, it is undesirable for an engine to possess a pronounced peak in its torque curve as it will then lack the flexibility and low-speed pulling which are especially necessary in a well-mannered road vehicle. It has been established by Farmer that a compound system – primary pipe, expansion chamber and tail pipe – works as if it were a plain pipe of equivalent (and much greater) length, but that the wave reflections are smoothed out by combination of the waves from the various components involved. This reduces the 'peaky' effect on torque, but there is still a chance that a 'bad' equivalent length will be selected, in which event the torque will drop at a certain speed range, increasing both above and below this critical period; as long as this is above the desired maximum obviously no harm is done and in fact it may provide a sort of automatic safeguard against a runaway engine. In the early motorcycle days it was common, largely because it was cheap, to fit a short pipe leading to a cylindrical expansion chamber with a small tail pipe, and this actually gives very good results when correctly proportioned. In deference to fashion many of these systems were discarded and replaced by long pipes leading to baffled silencers, almost always with deleterious results on the performance. There has since been a return on touring motorcycles to the short primary pipe, 12 to 16 in. long according to cylinder capacity, leading to a long tapering decorative silencer containing baffles which divide the space

into volumes of varying frequency and tailored by experiment to suit the particular engine concerned (Fig. 6.5). These give very good all-round performance and can be, though unfortunately not always are, very quiet. There is also a noticeable trend in competition engines to revert to the use of twin exhaust ports, each being fitted with its own

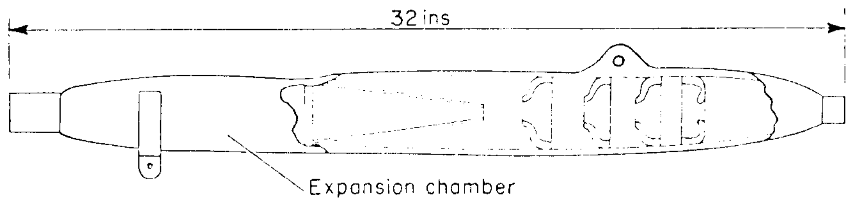

FIG. 6.5. The silencer of the 80 c.c. K11 Suzuki incorporates an expansion chamber in the front portion and four baffles in the rear portion.

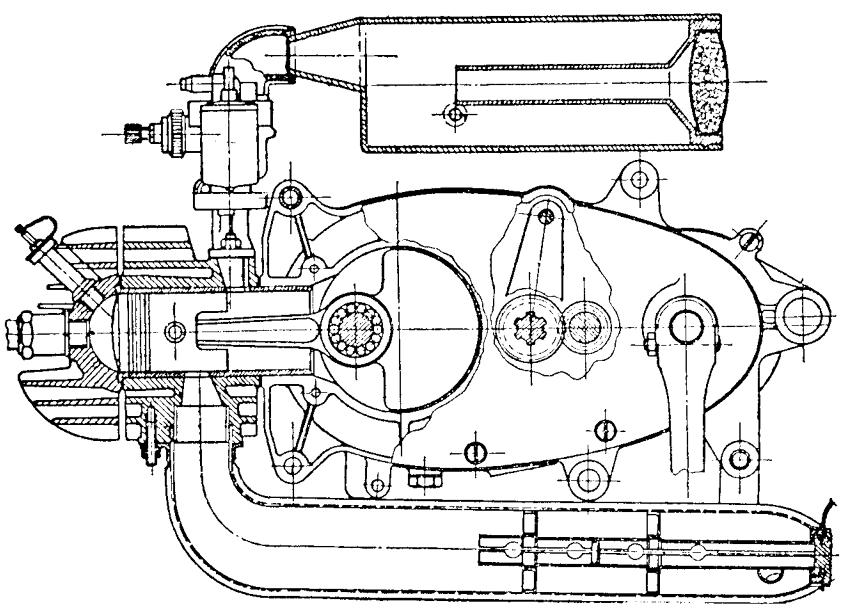

FIG. 6.6. The 'Jawa' 50 c.c. power-unit, showing the intake and exhaust silencers.

pipe and expansion chamber. Typical examples are the C.Z. (Fig. 12.14) and the Russian-built 'Kovrovets' (Fig. 12.15); in the latter, the pipes are curved upwards to help raise them well clear of the ground, a most important point in machines used for cross-country racing.

On scooters, where there is little length available and appearance beneath the cowling is immaterial, a box-like expansion chamber close up to the cylinder is used, sometimes with a plain tail pipe, sometimes

with a secondary expansion chamber of the absorption type with 'pepper-pot' tubes to absorb high-frequency noise, especially when four-stroking. Absorption silencers containing glass-wool or similar material will not remain effective for long, as they become clogged with oily residue. This by-product of petroil lubrication is a nuisance in any cool-running baffled system as it eventually builds up to a point where performance suffers, yet is difficult to remove without dismantling the silencer. Some examples are made demountable for easy cleaning, but unfortunately this is often taken as a means of removing the baffles, with an increase of noise but as often as not, a reduction in power.

Four-stroking in neutral, or worse still spasmodic firing on over-run, is particularly unwelcome on cars; fortunately there is plenty of room to house a bulky system containing two or three silencers, which are usually lagged to stamp out the metallic ringing caused by the sharp, erratic explosions. In such cases, care has to be taken to avoid fitting a system of 'bad' length.

At the other end of the scale, there is very little space available on chain-saws, but high specific power is required at 5 to 6,000 r.p.m.; the noise level is unimportant, but the emission of sparks may set off a conflagration costing millions. A common system is a conical megaphone, about 7 in. long, with perforated baffles to act as spark arresters, but the din emitted is so ferocious that it is actually painful to some operators, who work the machines for money rather than from choice. Actually the cutting action of the saw itself makes a considerable noise and this has been taken as an excuse not to bother with the exhaust. Recently more effective baffled silencers have been developed. On industrial and lawn-mower engines there is usually little space and the accent is on cheapness anyway; the usual solution is a short, blanked-off pipe, liberally drilled and surrounded by a canister of four or five times the cylinder volume, with a fish-tailed exit pipe. As they run hot, these not very efficient devices do have the merit of keeping themselves reasonably clean.

On outboard engines it is customary, except for some types of racing, to liberate the exhaust underwater, which itself acts as a very efficient silencer, although creating a pressure head of 15 to 20 inches of water. To overcome this, the outlet is placed in an area which, at speed, is under reduced pressure astern of the underwater gear; in the 'Mercury' the outlet is actually through the centre of the propeller boss. In order to avoid excessive peaks of pressure, the cylinder, or more often cylinders, exhaust directly into a large water-cooled expansion box formed in the main casting; in the six-cylinder Mercury units, this box is divided by a zig-zag wall so that only three cylinders exhaust into each half. This is necessary to avoid overlapping of the exhaust periods which would occur if more than three cylinders discharged into one chamber; even with three, there is about 20° overlap, but as the blast from one occurs

just as one of the other two is closing, and in fact the result may be beneficial as in the Saab pulse charging system. However, for racing, individual megaphone pipes, 17 in. long and tapering out to 4 in. diameter, are sometimess fitted to these engines and help to raise the r.p.m. from 5,800 to over 7,000.

A transom-mounted outboard does not vary its position relative to the water much and except in the more ebullient forms of aquatic sports the exhaust outlet is always submerged. When mounted as an auxiliary on a yacht this may not be the case; the outlet may regularly emerge as the boat rolls or pitches, and the resulting variation in back-pressure may cause the engine either to race or to stop, especially when cruising on very little throttle. In these conditions an overwater exhaust is less likely to hazard the ship and at least the sound level, even if noisier, is continuous instead of fluctuating widely as the vessel moves.

Safety Precautions

Engines for lighting sets or compressors are sometimes housed indoors in winter camps and the risk of poisoning the live inmates then becomes serious. Carbon monoxide is always present in two-stroke exhaust products and its effect is insidious and cumulative so that continual inhalation of only small amounts leads to illness which may be fatal; in fact Admiral Byrd nearly died from this cause in the Antarctic. Accordingly, the exhaust pipe must deliver to the outside air, all joints must be gas-tight and the system should be periodically inspected for leaks or rusty patches. However, extending the standard tail-pipe to a safe place out of doors may affect the power output or fuel consumption unless the extension is of large diameter, or else an expansion-box is interposed between the tail-pipe and the extension.

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NEW NISSAN SENTRA

Nissan's sweeping overhaul of its model line has finally trickled down to the entry level: For 1991 the Sentra gets an all-new replacement. Designed to vie in the hotly contested small-sedan category, the new Sentra's shape is conservative, but not unappealing.

The reborn Sentra comes with a choice of 16-valve DOHC fours in 1.6-liter 110-hp or two-liter 140-hp trim. Nissan focused development work on giving these engines broad powerbands while maintaining good fuel economy. The cars we drove both had prototype engines installed, with the larger one being much closer to the final production configuration. The difference showed: The revvy 1.6-liter engine, which features variable intake-cam timing and develops 108 lb.-ft. of torque at 4,000 rpm, buzzes a fair amount in the higher rpm ranges. By contrast, the all-aluminum two-liter engine generates silky, energetic pulling power throughout the rpm range, with its 132 lb.-ft. torque peak occurring at 4,800 rpm.

PREVIEW DRIVE

Finding power and tractability in an engine while keeping its appetite for fuel under control is a subtle business. According to Nissan engine designer Makoto Yasuda, major gains resulted from the development of remarkably long large-diameter intake runners with tapering interior diameters. This intake tract design efficiently boosts low- and medium-speed power by keeping intake-air velocity high and creating a strong resonance effect that helps ram the air-fuel mixture into the cylinders. With a five-speed manual transmission, the 1.6-liter engine earns EPA fuel economy estimates of 30 mpg in the city and 39 mpg on the highway; the two-liter motor rates 24 and 32 mpg, respectively.

Mated to a slick five-speed transmission in the SE-R performance version, the bigger engine makes for brisk acceleration and effortless cruising.—*Stuart F. Brown*

TWO-STROKE EXHAUST SYSTEMS

by

ROY BACON, A.M.I.Mech.E.

FIRST PUBLISHED January 1967
REPRINTED October 1967

76 Beckenham Road, Beckenham
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About the Author

Roy Bacon has long been a successful rider and helped found the Bantam Racing Club, being currently its Chairman. For three years he competed in the Isle of Man T.T. races using a Bantam, gaining a finisher's award each time and won the 1964 Bantam Club Championship for the best performance over the whole season. Apart from his ability as an organiser and rider, his tuning ability is without question and machines prepared by him have won over 200 awards. He is thus very aware of the problems confronting anyone trying to tune a two-stroke engine and this book will be found invaluable when tackling that complex task.

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CHAPTER ONE

Exhaust Pipe Systems

THERE are four basic types of exhaust system that may be used on a racing or high performance two-stroke engine. To a considerable extent, the type used will depend on the state of tune of the engine, the desired power characteristics and the use to which the engine is to be put. As the system becomes more complex to design and make, so it will have a more pronounced effect on the power output. By the use of the most complex system, the power output can be greatly enhanced.

While a greater power output can be obtained by the use of a sophisticated exhaust system, it must be remembered that unless the system is designed in relation to the engine, the end result may be disappointing. For this reason, it is often advisable to start experiments on the simpler systems and progress to the more complex as the complete engine and exhaust pipe are developed, and knowledge of the effect of the various pipes is gained.

Furthermore, the final use of the power unit must always be kept in mind. This is due to a general rule governing exhaust pipes that states that the more complex and effective the exhaust system, the narrower the power band will become, although the maximum power will be increased. This can only be taken as a general guide but is usually true; however, it

cannot be taken that the use of a complex system will automatically give a high power output, or the use of a simple one a wide power band. The use to which the power unit is to be put will govern the desired width of the power band if taken in conjunction with the number of speeds in the gearbox. Thus, racing engines tend to have complex exhaust systems and a large number of gears so that the engine can be kept at peak power all the time. On the other hand, engines used for moto-cross or karting need a much larger power band, in the first case due to the very variable ground over which the machine is to be ridden, and in the second due to the short straights and tight corners of most circuits used.

In this context, it should be noted that current works road racing machines employ between 6 to 12-speed gearboxes, the larger number of gears being used by the smaller capacity machines. Power bands of between 500-1,000 r.p.m. are usual with maximum power being developed around 12-14,000 r.p.m. Therefore, the type of power being developed by these machines is only useful if coupled with these multi-speed gearboxes. If the machine only has a 3 or 4-speed gearbox a much larger power spread is necessary otherwise the machine will lack acceleration when getting away at the start, and may lag after each gear change. Slipping the clutch can only partially overcome this as it wastes power and the clutch may not be able to withstand the abuse.

Before starting work on any exhaust system, the golden rules of the tuner must be well digested and never forgotten. These are: do one thing at a time and keep notes. Never alter more than one aspect of the exhaust system at a time, then try it and note every relevant factor. Not only does this include such things as maximum r.p.m., acceleration and speed at various fixed points on a circuit, but wind, weather and temperature can easily cloud the issue if allowed to. By carrying out the development work in the correct way, a pattern of information will result that can be of immense value when

dealing with the more complex systems. This may, on occasion, entail carrying out experiments using some unusual systems in order to obtain the extreme results between which it is hoped to work and also to obtain definite changes in results due to the alterations.

As few people have test bed facilities, field tests are usually the only way of determining the result of experimental work. Due to the day-to-day variations of wind and weather, it may prove necessary to carry out substantial changes each time to obtain a coherent pattern of results. Once this has been done, and the area which promises the best return found, further experiments can be conducted with much smaller incremental changes being made. If possible, these should be carried out on a single day during stable weather conditions, to ensure consistency of results. This demonstrates fully the necessity of only changing one thing at a time and of keeping careful notes on the results of all experiments, test results and development work carried out.

While carrying out this work, it must be remembered that the engine, and in particular the size, shape and direction of the ports play a great part in determining the maximum power output that can be obtained. In some cases, it may be found that changes to the exhaust system have little or no effect on the power output obtained. While this may be due to none of the systems being suitable, it can also easily be the engine itself that is not right and the trouble may be mechanical, thermal or thermo-dynamic in nature, or a combination of all three. It should also not be forgotten that as the exhaust system is altered, and becomes more sophisticated, it will, by its nature, require alterations to be carried out on the engine. At first, this may only entail modifications to the porting but as the power output is increased, it may become necessary to improve the mechanical design of the engine to cope with the greater power and increased r.p.m. This in turn may lead to changes in the thermo-dynamics and a reappraisal of the

thermal working may be required to deal with the extra heat generated. Thus, changes in the exhaust system can in time mean considerable changes to the complete engine due to the increase in power and engine speed.

When considering the exhaust system, it is most important that the size, shape and length of the exhaust port is taken into consideration as it is, of course, part of the system and can affect results if ignored.

The four types of exhaust system that may be used are as follows:

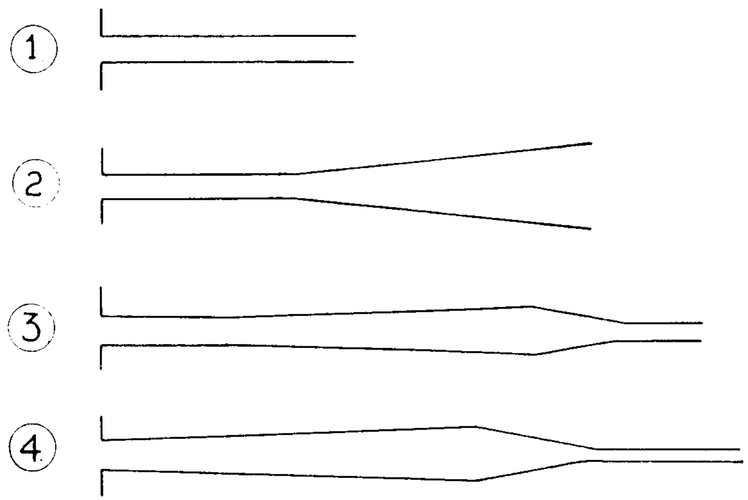
1. Plain pipe of constant diameter cut off square to the desired length. Thus, length and diameter are the only variables.
2. Plain pipe of constant diameter with megaphone of constant angle of taper attached to the end furthest from the exhaust port. Note that the length of plain pipe could be zero in which case the megaphone would start at the exhaust port. This, in fact, gives a system that is the same as the first part of type 4 listed below. The variables of the system have now increased to diameter and length of the pipe, and angle of taper and length of megaphone. It should be noted that the smaller diameter of the megaphone is the same as that of the pipe, and that the larger diameter is dependent on the angle of taper and the length of the megaphone.
3. Plain pipe with expansion box fitted to the end furthest from the exhaust port. The expansion box may be of any desired shape but usually comprises a shallow tapered divergent megaphone to the end of which is attached a short tapered convergent megaphone which reduces the exhaust system diameter again. Attached to this may be a tailpipe of constant diameter.

   With this system the number of variables has further increased to diameter and length of the plain pipe, angle of taper and length of first megaphone, length and smaller diameter of second megaphone, and length and diameter

   of tailpipe. It should be noted that normally the larger diameters of the megaphones are the same, and the smaller diameter of the second megaphone and that of the tailpipe are the same. Usually this diameter is less than that of the exhaust pipe. The working of the whole system is partly dependent on the total length of the pipe as well as those of the constituent parts so that the system is much more difficult to experiment with than the first two described above.
4. Fully tapered expansion box or resonant pipe. This is the same as type 3 above except that the pipe expands from the exhaust port. Thus the first part of the pipe comprises a shallow tapered divergent megaphone attached directly to the exhaust port of the cylinder barrel. The second part is a tapered convergent megaphone and to this is attached a tailpipe. Thus, the variables for this system comprise angle of taper and length of the first megaphone, length and smaller diameter of second megaphone and length of tailpipe if its diameter is assumed to be the same as the smaller diameter of the second megaphone. Once again the tailpipe diameter is usually much smaller than the exhaust pipe diameter, or in this case, the diameter of the system at the junction of the pipe to the cylinder barrel. It should be noted that in all cases, and in particular this one that the exhaust system starts at the exhaust port in the cylinder barrel. Therefore, when a resonant exhaust pipe is in use, the port should diverge to the same degree as the first megaphone.

The four exhaust systems are shown in Fig. 1 as straight systems before being bent to form to suit the machine. This form does partly depend on the size of the system and the use to which the machine is to be put. Thus road racing machines usually have the pipe tucked in under the engine to ensure that adequate ground clearance will still be available when the machine is banked over and the suspension compressed. Moto-

cross machines usually have the system higher but still well tucked in due to the larger vertical ground clearance needed, the existence of water splashes and thick mud on some circuits, and to avoid damaging the system on trees or rocks. Karts have the system installed to fit in the space available with adequate ground clearance and ample protection from accidental damage.



(Fig. 1) *The four basic forms of exhaust pipe shown before being bent to shape to fit the machine. (i) plain pipe, (ii) pipe and megaphone, (iii) pipe and expansion box, (iv) resonant pipe.*

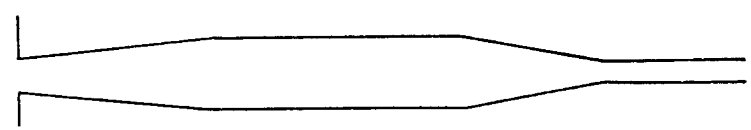
Due to the restricted space available, installation on a Kart may be particularly difficult as normally a silencer will have to be added to the exhaust system which takes up even more space. By careful tuning, the addition of this silencer to a resonant exhaust system should not affect the power output but will ensure that the machine complies with the silencing regulations governing Karts.

Of the four types listed above, the last gives by far the best results when applied to a suitable engine that has the carburation correctly adjusted. However, while it will assist in pro-

ducing a considerable increase in power, it does so only at the expense of narrowing the power band and requiring the engine to be more carefully tuned, assembled and maintained. Thus, as with any piece of precision mechanism, it must be correctly treated if the best results are to be obtained. With this type of exhaust system, it is imperative that such things as ignition timing, piston clearances, petroil ratio and type of plug are closely controlled. If they are not the results are likely to be variable or disappointing.

The first three systems listed above are relatively easy to make or modify. The last is far more difficult from both aspects due, usually, to the need to bend the first megaphone portion. The exception to this is when a rearward facing exhaust port is used as this allows a substantially straight system to be fitted to the machine, bends only being needed to avoid frame members and the rider's legs.

Due to the difficulties that arise in making and modifying a resonant pipe, the shape may be changed to that shown in Fig. 2 as this allows experiments to be carried out more easily. The same shape may also be used for an expansion box used in conjunction with a plain pipe. With this form of expansion chamber, it is far easier to alter the volume by changing the length of the cylindrical portion either by cutting and welding or by the use of a slip joint. The tailpipe may also be modified in length by similar means.



(Fig. 2) *Modified chamber shape used during experiments.*

Without the cylindrical portion, it is a far more lengthy process to experiment with alternative systems as it is necessary to alter the larger diameter of both megaphones if it is desired to change the volume of the pipe. If the diameter should be

fixed it would then become necessary to alter the angle of taper of one or both megaphones.

In a similar manner, experiments may be carried out on the tailpipe diameter by welding a washer inside the pipe and drilling it out in stages. This produces a change in the restrictive effect of the tailpipe which will affect the action of the system as a whole.

Further experiments may be carried out by using special flexible tubing which may be stretched to up to twice its normal length. With this incorporated in the system, the length and volume of the exhaust system may be altered during a race but it should be remembered that due to the internal corrugations of the tubing the effective bore is reduced.

With all these expedients, it should be remembered that one of the prime objects of the system is for it to place as little restriction as possible on the passage of the exhaust gas so that when the experiments have been concluded, the data collected should be analysed and used to determine the shape and dimensions of the desired exhaust system.

Despite the point mentioned above, it may well be found that the system that is theoretically the best shape gives poorer results than the temporary arrangements used during experiments. If this is so, it may be due to one of two factors, or both of these factors. The first is that the interpretation of the accumulated data may not be correct and a reappraisal of the information available may enable the reason for the reduction in performance to be seen. The second is that the fault may lie in the engine and not in the exhaust system. In this case, the experimental exhaust pipe may be the best for the engine as it was during the tests but the engine layout may not be the best for the new system. Once again it can be seen that, if fully detailed notes have been taken of all stages of engine development, it is much easier to decide which factor is giving rise to the poorer results and the best remedy to take.

The desired effect of the various types of exhaust pipe is to

increase the maximum power, or to obtain a reasonable power output with good mid-speed range torque, depending on the use to which the machine is to be put and the number of speeds in the gearbox. As this effect is based on the complex action of the waves in the pipe which is further complicated by the continually changing conditions in the engine, each cylinder must have its own system. If more than one cylinder was connected to a single system, the inter-action of wave pulses between the two cylinders would make the already complex situation, even more confused, especially as minute differences in the cylinders would have a marked effect on the form of the gas waves. This could arise from such things as slightly different exhaust port timings, port shapes, length of pipe joining cylinder to common system, or small variations in gas temperature.

Due to these considerations, all high performance two-strokes must be fitted with a separate exhaust system to each cylinder. As is normal with multi-cylinder engines, each cylinder and all the parts that affect it such as carburettor and ignition as well as the exhaust system should be matched as nearly as possible. In a few cases, mainly in the moto-cross field, single cylinder machines have been fitted with two exhaust systems. These two systems are matched and may be fitted for one of two reasons. The first is when a bridged exhaust port is used and is based on the theory that once the exhaust gas has been split into two streams it is better to allow them to continue into two separate exhaust pipes rather than to join them together again with the possibility of turbulence occurring at the junction.

The second reason is the more simple one of installation. With the rather bulky systems favoured on modern moto-cross machines, it is very difficult to fit a single pipe out of the way. If too low, so that it can be tucked in, it may easily be damaged by rocks as they reduce the ground clearance. If fitted high up, it may be in the way of the rider's leg. With

two smaller section pipes, these can be mounted high up, well away from the ground, and be tucked in out of the way.

The use of twin exhausts on a single cylinder machine has not been used in the road racing field to any great extent, as the ground clearance problem is not so acute, the pipe only having to be well tucked in. In the moto-cross field, the recent tendency has been towards the use of smaller systems so that a number of machines that used to have twin exhausts are now fitted with a single exhaust system. Despite any theoretical advantages, it is generally agreed that a single exhaust system gives a higher power output and more torque than a twin pipe, so that the majority of high performance two-strokes are fitted with a single exhaust system to each cylinder and each exhaust system is kept separate from any others.

Therefore, all the following notes are based on a single cylinder engine fitted with a single exhaust system. The case of a multi-cylinder engine is simply a case of duplicating the appropriate size and shape of system, while a single fitted with a twin exhaust system can be considered as one with the length of either system but the volume of the two taken together. Thus, the last case can be thought of as a system of the same length as one pipe but with twice the cross-sectional area of one pipe at any point. This is not quite correct due to the difference in megaphone angles that arise between the comparisons but is a reasonable assumption that allows comparisions to be made between single cylinder engines fitted with single or twin exhaust systems.

CHAPTER TWO

Exhaust Pipe Theory and Practice

THE whole purpose of the various forms of exhaust system is to increase the power output of the engine over a certain speed range by improving the volumetric efficiency. While this may reduce this efficiency at other engine speeds, depending on the characteristics of the system, this is the penalty that has to be paid to obtain the desired high power outputs.

The power generated by the engine in the cylinder is known as the Indicated Horse Power and

$$\text{I.H.P.} = \frac{\text{PLAN}}{33{,}000}$$

where P = mean effective pressure, lb/in^2
L = stroke, feet
A = piston area, in^2
N = power strokes per minute.

The power output is known as the Brake Horse Power and

$$\frac{\text{B.H.P.}}{\text{I.H.P.}} \times 100\% = \text{Mechanical efficiency.}$$

Thus the power output of a given size of engine is mainly dependent on the mean effective pressure and the engine speed. The M.E.P. is mainly dependent on the compression ratio and the quantity of mixture contained in the cylinder and as the

compression ratio is determined by the fuel being used, and so is fixed, the power is dependent on the M.E.P. and thus on the breathing of the engine.

This breathing is greatly affected by the exhaust system and use is made of the energy in the exhaust gas to assist the cylinder filling. Naturally the breathing is also affected by the inlet pipe and transfer port characteristics but while these can play a considerable part in developing a high power output, it is the exhaust system which may have the greatest effect when the highest power output is desired. Thus a 125 cc. engine developing 5 b.h.p. may be tuned to deliver 9 b.h.p. by relatively simple changes to the induction and porting but still fitted with a simple exhaust system. However, the power may be boosted to 15 b.h.p. by the use of a more sophisticated exhaust pipe while still retaining basically similar inlet and transfer ports and times.

The increase in power is brought about by the action of the pressure waves in the exhaust gas. These act as follows. When the exhaust port opens the cylinder pressure is still above that in the pipe as this is connected to atmosphere. Due to the pressure difference that exists, the gas flows from the cylinder and a pressure pulse is caused to pass through the gas. When this pulse reaches the end of the pipe, it is reflected back toward the cylinder as a scavenging wave. When this wave reaches the port, and assuming that the port is still open, it leads to a further scavenging of the cylinder. This gives a marked drop in cylinder pressure.

The scavenging wave is reflected away from the cylinder and as the port is now partly closed the wave is reflected in both positive and negative senses. As it has also lost some energy due to wall friction and end losses, its amplitude and therefore the work it can do, are decreased. Thus it is the primary wave that is mainly used to do useful work.

Due to the drop in cylinder pressure, a pressure gradient is created across the transfer ports and this in turn initiates a

scavenging or rarefaction wave in the transfer ports. This wave passes towards the crankcase while the gas is flowing to the cylinder. When it reaches the end of the transfer passage, which may be considered as an open ended pipe, it changes sign and returns as a ramming wave to the cylinder. Depending on the state of opening of the ports, it may be reflected again either with or without change of sign and these wave trains may superimpose with those in the exhause pipe to form a complex but stable wave formation throughout the engine.

From the above it can easily be seen that by combining the appropriate wave actions with port timings a great increase in power is possible. It is desirable that the wave in the exhaust pipe is used to evacuate the cylinder of the burnt charge and draw the fresh charge up into the cylinder, this charge being assisted by the ramming effect of the induced waves in the transfer ports. Therefore, the transfer ports should be shut when this ramming effect has reached a maximum and the exhaust port shut before the fresh charge, under the stimulus of the pressure generated by this ramming and the scavenging effect of the exhaust pipe waves, can escape into the exhaust system. This is prevented by a positive pressure wave that is generated in the exhaust system in one of two ways detailed more fully below.

While the aim of the exhaust system may be expressed in simple terms, the derivation of a suitable system for any one engine is a highly complex business that entails considerable development even when some of the basic points can be calculated. As these are mainly theoretical, they can only act as a guide to give a suitable starting point for the system which will then need developing to its highest pitch. Because of this and due to the effect of other factors, such as inlet and transfer port sizes and timings, it is not possible to lay down a series of simple formulæ for calculating the sizes of pipes, expansion boxes or resonant chambers. However, it is possible to give some guidance as to the best avenues to explore, many of these

having been confirmed by experimental work and practical applications.

If a plain pipe is exhausting directly to the atmosphere, the outward pressure pulse is reflected instantaneously. Due to the sudden reduction in pressure that occurs the gas particle velocity is doubled and therefore a fourfold increase in kinetic energy occurs and this is lost to atmosphere instead of being used to assist the cylinder filling. If a megaphone is used on the end of the pipe, the process of reflection is gradual and the particle velocity is reduced. This ensures that the bulk of the gas energy is transmitted back to the port so that a greater power output is possible. In addition, the duration of the reflected pulse may be extended so that the engine speed becomes less critical over the range that the pulse operates over. Outside this range the effect of the megaphone may be detrimental but this is of no concern with a high performance two-stroke.

The effect of the pulse is dependent on its passage up and down the pipe and the speed at which it travels depends on the speed of sound in the gas. This varies considerably, depending on the temperature and speed of the gas, from 1,000ft/second for an inlet port to 4,000ft/second at the exhaust port. Thus a figure of 3,000ft/second may be assumed a reasonable average for the first 12in. (305mm). This is used for calculations of a plain pipe, pipe and megaphone and pipe and expansion box, but only for the pipe portion of the system. The remainder of the system is likely to have an acoustic speed of around 1,500ft/second so that this is the figure used. A resonant chamber may have a slightly higher average acoustical speed over the first megaphone portion of around 1,800ft/second, dropping to 1,500ft/second for the rest of the chamber.

As the pulse speed is a constant it follows that if the engine speed changes the time that the ports are open also varies so that the peak values of the pulse only reach the correct points in the engine at any one engine speed. This effect is less appar-

ent when a megaphone is fitted to a plain pipe due to the more gradual wave reflection that takes place. It is apparent from the above that as the engine speed increases the pipe length has to be shortened as there is less time between the instant the port starts to open and when it finally closes. Also as the port timing is extended so a longer pipe is needed as the time for the wave to travel in the pipe is extended if the engine speed remains constant.

Mention has been made of a positive pressure wave that is generated in the system to prevent loss of fresh charge. With a plain pipe this occurs after the wave has travelled up and down the pipe twice. This occurs as the wave starts positive, is reflected as negative at the open pipe end, is reflected still negative at the cylinder due to the cylinder pressure still existing acting as a closed end to the pipe, and is then reflected as positive at the open end to produce a positive pressure at the port. Thus during the time the port is open the wave has to cover a distance equal to four times the length of the pipe.

This length may be calculated for a plain pipe as follows:

The period that the port is open $= \dfrac{\theta}{6N}$ seconds.

Where θ = total opening period, degrees
N = engine speed, r.p.m.
and this time should equal the time it takes the pressure pulse to travel up and down the pipe twice.

Therefore $\dfrac{\theta}{6N} = \dfrac{4L}{12a}$ where L = pipe length, inches.

a = acoustic velocity=1800ft/sec.

Thus $L = \dfrac{\theta a}{2N} = \dfrac{\theta}{N} \cdot 900$

This illustrates the interaction between the pipe length, exhaust port timing and engine speed.

From this equation, the following table may be drawn up to show values of L for various engine speeds and port angles.

[rpm]

θ	6000	7000	8000	9000	10,000	11,000	12,000
120	18	15·4	13·5	12	10·8	9·8	9
130	19·5	16·8	14·7	13	11·7	10·6	9·8
140	21	18	15·8	14	12·6	11·5	10·5
150	22·5	19·3	16·9	15	13·5	12·2	11·2
160	24	20·6	18	16	14·4	13·1	12

The above table shows the small variations in correct lengths for changes in the two variables and emphasises the need for care when experimenting with different lengths of exhaust system.

When a pipe is fitted with a megaphone, the more gradually reflected negative pulse is itself reflected as a positive pulse from the open end and this returns to the port to prevent loss of the fresh charge. In order that the fresh charge is subject to the maximum depression without loss it is therefore desirable that the negative pulse should arrive at the exhaust port when the transfer ports begin to open while the positive reflection should arrive when the exhaust port is on the point of closing. Thus the length of the pipe is determined by the blowdown period and the total length of the system is obtained as above but allowing for the pulse to travel up and down the system once only. Therefore the formula becomes:

$$\frac{\theta}{6N} = \frac{2L}{12a} \text{ and } a = 1500\text{ft/second average.}$$

$$\text{Thus } L = \frac{\theta a}{N} = \frac{\theta}{N} \cdot 1500$$

and therefore the total length of pipe and megaphone is nearly twice that of the pipe alone.

The pipe length is dependent on the blowdown period and the engine speed as follows. B = Blowdown period, degrees.

$$\frac{B}{6N} = \frac{2Lp}{12a} \text{ where } a = 3000\text{ft/second}$$

$$\therefore Lp = \frac{Ba}{N} = \frac{B}{N} \cdot 3000$$

The following table shows some pipe lengths for different blowdown periods and engine speeds.

[rpm]

B	6000	7000	8000	9000	10,000	11,000	12,000
10	5	4·3	3·8	3·3	3·0	2·7	2·5
15	7·5	6·4	5·6	5·0	4·5	4·1	3·7
20	10	8·6	7·5	6·7	6·0	5·4	5·0
25	12·5	10·7	9·4	8·3	7·5	6·8	6·2

As well as requiring the pressure pulse to reach the port at the right moment, it is also desirable that it contains as much energy as possible. Therefore, a pulse of high amplitude is needed. It has been shown that this may be achieved by having a high rate of exhaust port opening. This is achieved in two ways. Firstly the top of the port is made square with the axis of the barrel so that the entire width of the port is opened at once. Secondly, the port height is kept down and the width increased as much as possible. The latter requirement is more difficult to attain as such factors as blowdown period, practical port width, transfer port timing and size, and the breathing of the engine at maximum r.p.m. all have a bearing on the situation.

It can further be shown that a small diameter exhaust pipe gives a better theoretical air consumption curve than a large one. However, this has to be balanced against the greatly increased frictional losses that arise with the smaller pipe so that in this case the pipe size has to be a compromise based on the exhaust port area.

To further augment the power output of the engine either an expansion box may be fitted to the exhaust pipe or a resonant chamber may be used. These will produce more power at the expense of critical carburation and a narrow power band as the maximum torque and power occur at nearly the same engine speed. This is particularly true of the resonant chamber. The action of the reverse cone is to provide a closed end pipe effect of a gradual nature so that the initial negative pressure wave is reflected as a positive wave that returns to the exhaust port. Thus the cylinder is scavenged by the negative pulse, which also initiates a ramming wave in the transfer passages, and the fresh charge is allowed to spill into the exhaust system but is forced back into the cylinder by the returning pressure wave. As the first wave cycle is the one of greatest amplitude, it produces the maximum effect if used correctly as it contains the most energy. Subsequent waves contain less energy due to attenuation of the wave.

Naturally, the diameter and to some extent the length, of the tailpipe play a considerable part in this effect as the degree of restriction will affect the pressure build-up in the chamber and this in turn affects the speed at which the positive wave returns to the exhaust port.

From the above it can be seen that the wave action in the resonant chamber is very complex in nature.

However, some guidance may be obtained from the above notes and certain basic assumptions made. The desired effect of the resonant pipe is to increase the volumetric efficiency of the engine and so raise the mean effective pressure and the addition of the reverse cone and tailpipe to the plain megaphone make the effect far more pronounced.

The engine speed at which maximum torque will be developed is dependent on the length of the tapered section and on the length and diameter of the tailpipe. It is also affected by the volume of the chamber. A general guiding rule may be stated that as the engine speed for maximum torque increases

so the length and volume decrease, while the tailpipe length increases and its diameter reduces. As these points do interact this can only be taken as a general guide.

Therefore, it is usual for a moto-cross engine to be fitted with a large volume resonant pipe that terminates in a short tailpipe whose diameter is nearly as large as that of the main megaphone at the exhaust port. Conversely, road racing engines tend to have small chambers and long tailpipes of small exit diameter. Between these two extremes all combinations may be used to obtain the best results for a particular engine.

As the length of the tailpipe has an effect on the maximum r.p.m. the engine will reach, a compromise is often made. This is done by using a large chamber and large bore tailpipe to keep the r.p.m. at which maximum torque occurs as low as possible, but to use a long tailpipe to obtain at high maximum speed. Often this compromise is modified to suit the circuit being raced at, a short pipe being used on a twisty circuit and a long one at a fast circuit where top speed is at a greater premium than maximum acceleration.

It is impossible to give specific dimensions for any particular engines but the following notes give guidance as to the proportions of the various systems that may be used on two-stroke engines. It must be emphasised that the figures given are only a guide and must not be taken as the best for any one engine. They only serve to give a starting point from which further development may be taken to achieve optimum results.

The plain pipe is the simplest form of exhaust system and once the diameter has been chosen, the length may be arrived at by cutting the pipe down in stages. The pipe should not be shortened by too large an amount at a time, ½in. (12.7mm) is a reasonable figure. As an alternative to cutting, a sleeve can be made to slide along the pipe to vary the length. This allows the approximate length to be determined, after which the pipe can be made to this plus an inch, and experiments continued by sawing ·2in. (5mm) off at a time.

The internal diameter of the exhaust pipe may be taken as laying between ·6 - ·7 of the bore of the engine, and the length as between 6 - 8·5 times the bore of the engine. Normally the higher the engine speed the shorter the pipe. Thus if the engine peaks around 6,000 r.p.m. a factor of 7·5 or 8 may be used, but if it peaks at 7,000 r.p.m., this would drop to between 6·5 and 7.

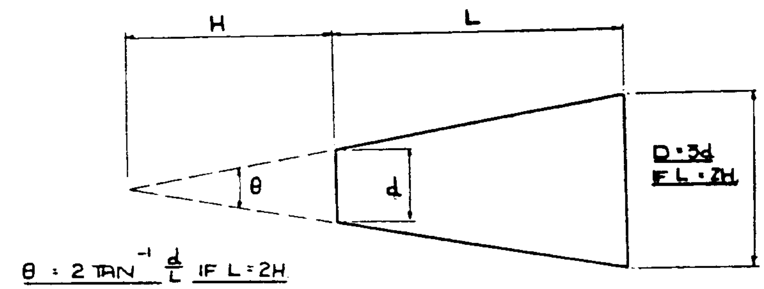
Thus a 125 cc. cylinder which peaks at 7,000 r.p.m. and has bore and stroke equal at 54mm would need a pipe of between 1¼in. - 1½in. diameter and about 13½in. - 14¾in. long. Note that this length includes that of the exhaust port so that if, in this case, the port is 2¼in. long from cylinder to pipe mounting face, the pipe length would lay between 11¼in. - 12½in.

Similarly, a 250 cc. cylinder might have an exhaust pipe of between 1⅝in. - 1⅞in. diameter and between 16in. - 23in. long depending on peak r.p.m. A 50 cc. cylinder would be expected to have pipe dimensions of $\frac{15}{16}$in. - 1⅛in. diameter and 9½in. - 13½in. long. Due to its small size, the 50 cc. cylinder would be likely to use the larger diameter and the shorter length.

The above notes apply also to a plain pipe and megaphone exhaust. In this case, the pipe would follow similar lines both in dimensions and method of tuning. The megaphone length would be about the same as the pipe length and should be formed so that the length of megaphone is equal to twice the distance from the start of the megaphone to the apex of the megaphone cone. This is shown in Fig. 3. From this it can be shown that D=3d and that the included angle = 2 Tan $^{-1}$

$$\frac{d}{L} \text{ as } \frac{L}{H} = 2$$

This angle may be expected to lay between 6° - 12° so that the following table may be constructed assuming that L/H=2. This figure is chosen as it provides the best compromise between preventing energy loss due to a too abrupt pipe end and gas friction losses. With higher values the friction losses out-



(Fig. 3) *Basic form of megaphone.*

weigh any gains while if the value is reduced to unity, there is a loss due to the prolonged presence of a stationary shock wave at the junction of the pipe and the megaphone.

In the table below, the pipe diameter is taken as unity, other values of length being obtained by multiplication.

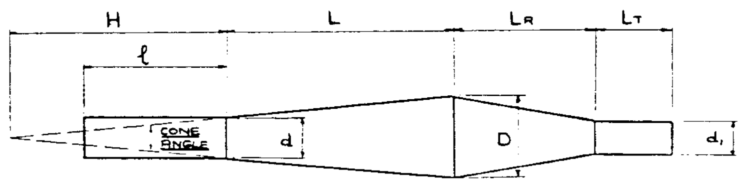
$\theta°$	6	7	8	9	10	11	12
Lin.	19	16·3	14·2	12·6	11·3	10·3	9·4

Thus with an included angle of 10° the megaphone length would be 11·3in. for a 1in. diameter pipe and 15·5in. for a 1⅜in. diameter pipe while the megaphone exit diameters would be 3in. and 4⅛in. respectively.

The expansion box attached to the end of a plain pipe is also relatively easy to experiment with. Once the size of the box has been chosen, the length of pipe it is attached to, and the length of the tailpipe may easily be changed. The chamber volume may be altered by adding a cylindrical portion at the junction of the two megaphone cones. Suggested sizes for the pipe are a diameter of ·6 - ·7 bore and length of 6 - 8·5 bore with preference being given to the shorter length. The first cone is likely to increase its diameter to 1·4 - 1·8 bore in a length of 8 - 10 bore. The reverse cone will reduce the diameter to ·5 - ·6 bore in a length of 3 - 4 bore and have a tailpipe length of 1 - 3 bore. The tailpipe diameter is invariably less than the plain pipe diameter. From these figures it can be

determined that the cone angle will be 4° - 6° and the value of L/H will lay between 1·0 - 1·5. The following table shows suggested sizes for various size cylinders. The letters refer to those shown in Fig. 4.

Capacity	Bore	d	D	d_1	l	L	L_R	L_T	Cone Angle	$\frac{L}{H}$
cc.	mm.	in.	in.	in.	in.	in.	in.	in.		
50	40	·94	2·25	·8	9·0	13·0	5·0	4·5	5°30′	1·25
125	54	1·28	3·0	1·12	15·0	19·0	7·0	4·0	5°10′	1·34
250	68	1·6	4·0	1·4	20·0	24·0	8·5	5·0	5°44′	1·5



(Fig. 4) *Expansion box and plain pipe in basic form.*

As noted above, the tailpipe length is mainly dependent on the maximum speed of the engine so will vary considerably depending on the shape of the power band.

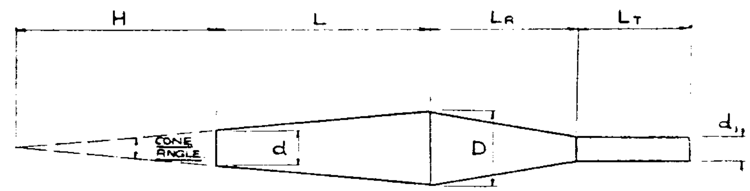
The resonant chamber exhaust system is far more difficult to experiment with than the types above as only the tailpipe length lends itself to easy changes. The volume of the chamber may be altered as above but if the cone angle is changed, a new system has to be made.

The dimensions follow a similar pattern to those of the expansion box suitably modified to cater for maximum torque at a higher engine speed and a higher value for the maximum engine speed. Suggested sizes are small diameter of first cone ·6 - ·7 bore, large diameter 1·4 - 1·6 bore, length of first cone 11 - 15 bore. Second cone length is 3 - 5 bore and its smaller diameter ·4 - ·6 bore, or ·75 - 1·0 of first cone small diameter. Tailpipe length may be 1 - 5 bore depending on the maximum

engine speed. These figures produce a cone angle of 3° - 4° and L/H factor of 1·5 - 2·0.

Referring to Fig. 5, the following table may be drawn up for various cylinders.

Capacity	Bore	d	D	d_1	L	L_R	L_T	Cone Angle	$\frac{L}{H}$
cc.	mm.	in.	in.	in.	in.	in.	in.		
50	40	·94	2·5	·62	23	6·5	7·5	3°54′	1·66
125	54	1·28	3·25	1·0	28	7·5	8·0	4° 2′	1·54
250	68	1·6	4·25	1·4	35	9·0	7·0	4°20′	1·65



(Fig. 5) *Resonant pipe details.*

Once again it must be stated that these figures are only a guide and that the dimensions chosen have to take into account the characteristics of the engine and the use to which it is put. It must also be remembered that the use of a resonant exhaust system will materially affect the running of the engine. It is likely to increase the operating temperature and so call for a larger main jet and may call for a shorter exhaust lead due to the increased efficiency of the blowdown period.

CHAPTER THREE

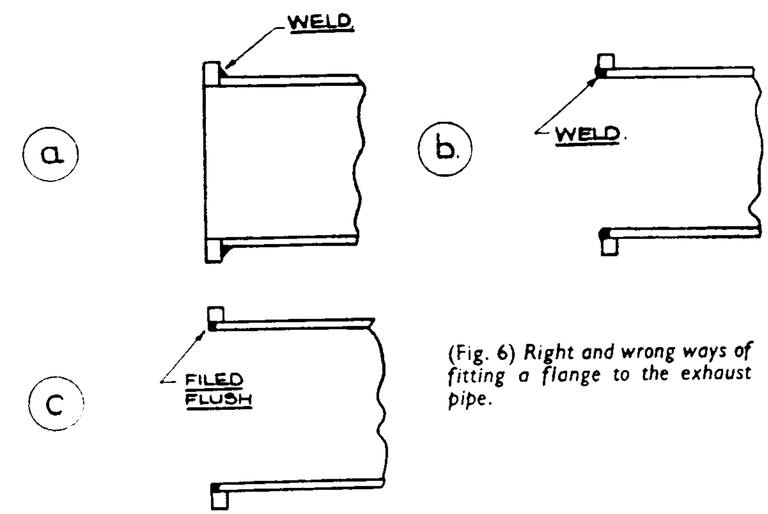
Manufacture, Materials and Finish

ONCE the shape of the exhaust system has been decided on, it then has to be made. The degree of difficulty involved in this depends on the type of pipe used, the more useful shapes also being the hardest to make.

Whichever type is to be made, it will normally require a flange and gasket at the exhaust port end except where a slip joint of the type described in the following chapter is used. In this case, only a sleeve is required on the port end of the pipe. When a flange is used, it may be rolled out of the pipe material or be a separate steel washer welded on. This should be done as shown in Fig. 6 to ensure that the flange nut has a good shoulder to pull up against. Note that in (a) the weld prevents the flange nut from pulling up square and there is no guarantee that the flange and pipe will be in line. (b) and (c) show how this is overcome to produce a smooth transition from port to pipe and a stronger joint.

N.B.—*With the more complex systems the flange nut must be fitted to the pipe before the flange is welded in place.*

The gasket should be located radially and is normally a copper or aluminium ring. While the more usual copper-



(Fig. 6) *Right and wrong ways of fitting a flange to the exhaust pipe.*

asbestos type may be used, it must be watched due to the gasket compression that takes place. As this may allow the flange nut to become loose, this type of gasket cannot be recommended, the solid type is much better.

The plain pipe type of exhaust system is easy to make, as it only consists of a piece of tubing bent to the required shape, cut to length and fitted with a flange. Often an existing exhaust pipe may be cut down to the required length and used without further modification.

The plain pipe and megaphone is not much harder to make as the pipe portion remains the same so that only the megaphone part has to be added. This is usually made by rolling a sheet of metal into a conical form and butt welding along the join. A short sleeve is added and part slotted, so that it may be pushed onto the end of the pipe and clipped in place. Alternatively, it may be welded to the straight pipe.

The case of the pipe with expansion box is again fairly easy

as it is only an extension of the above procedure. In this case, however, it is necessary to roll up two cones, weld them together, add a tailpipe and then attach the assembly to the plain pipe.

With the three types above the problem of bending the pipe is greatly simplified by its constant diameter even if a standard pipe cannot be modified. The last type, the resonant pipe, is a far more difficult proposition, as no standard pipes that could be modified are available and it is not easy to bend a tapered pipe. While the bent portion can be made from two pressings welded together, this is only practical if the quantities involved warrant the cost of the tools. Normally, they do not, as each exhaust system is individual to the engine it is fitted to.

The general form of the pipe is not difficult to produce as once again it only involves the rolling of two cones, welding them together, and adding a tailpipe. This only leaves the problem of bending.

The pipe can be bent hot if filled and normally the first megaphone is bent before being welded to the other parts. It is also usual to provide an extra 2in. (50mm) at each end so that the pipe can be held during the bending operation. Once this is done, it is cut to length and it and the other parts welded together.

Whichever type of exhaust system is being made, it is important that all bends are smooth and without ripples. All joints must be smoothly welded on the inside so that the gas flow is not disturbed. While this applies to all types, it is particularly important when dealing with resonant pipes and with most of these, the first 12in. (300mm) or so must be smooth. Certain liberties may be taken with the form of the rest of the system to enable it to be fitted to the frame, but naturally these should be kept to a minimum if possible.

It is difficult to bend a resonant pipe so that it matches up with the exhaust port but this may be partly overcome by

making a short stub with flange to attach to the port and mating the pipe with this. In most cases this is easier to do as the problem of ensuring that the flange is hard against the port is removed. In addition the flange nut can be added at the final joining stage which makes checking the flange to port joint easier.

The support brackets should not be fitted until the pipe has been attached to the exhaust port. Once this has been done, the brackets can be fitted and welded to the pipe and the frame. The final operation is normally drilling through both parts for the fixing bolt. This should be carried out with the weight of the pipe supported by a stand.

The exhaust system is usually made of mild steel although where a separate megaphone is used, this may be made in aluminium. If this is done any support should be riveted in place with a number of rivets to prevent it pulling away, a prevalent fault with this material.

The bent part of the exhaust system should be made from ·064in. (1·6mm) steel tubing or sheet as this allows the bends to be formed more easily. The straight portion may be ·036in. (0·9mm) steel to reduce the total weight of the system. Some resonant pipes have been made entirely from the thinner material but the bending operation becomes even more tricky than usual. If it can be accomplished however, it does gives a worthwhile weight saving of about 2lb. If the first 15in. (381mm) is made in the thicker material, the weight of the pipe is increased by 10oz. Thus lightening the system has to be balanced against the ease of manufacture.

Once the system has been made, it may be left unfinished or a protective coat added. As it is desirable that the system should radiate the exhaust heat it should not be chrome plated. This is because a dark surface will radiate heat much better than a light or shiny one. The best finish is matt black and this may be obtained by painting with a heat resistant paint or by one of the chemical finishes that are available. As

it is likely that the system will be frequently modified, to carry out experiments, in most cases it is best left plain and lightly oiled to prevent rust if left for long periods. Most systems become covered in oil during use due to drips from the engine and gearbox and as the castor base oils seem to produce an excellent rustproof black finish if baked on by the heat of the pipe the finish may usually be left to look after itself.

CHAPTER FOUR

Mounting the Exhaust System

If the exhaust system is functioning correctly, the restriction it places on the passage of the exhaust gas should be a minimum and so, in theory, it should be straight. Due to the wave action taking place in the system, a straight pipe might not give the optimum result as this result could depend on the need of a small restrictive effect in the pipe. However, this would be due to the system not being absolutely correct for the engine and its performance band rather than for the need of an artificial restriction.

In practice, this is of somewhat academic interest, as at least one change of direction in the path of the gases is invariably necessary, and sometimes more. Therefore, the system has to be developed with the installation in mind and due allowance made for the changes in direction that the gases will be subjected to. As the gas will lose energy whenever it is forced to change direction and it is desirable that the gas energy is used to charge the cylinder and not be dissipated unnecessarily, the exhaust system should have the minimum number of bends in it.

When the pipe has to be bent, this should be gradual in nature and of the maximum radius that is practical. The bend or bends should be smooth and any changes in section that

occur in the more complex systems should not be too abrupt. Thus the junction points of divergent and convergent megaphones or megaphone and tailpipe should be clean internally without any welding lumps protruding into the gas stream as these will create local pockets of turbulence. If this turbulence is excessive, it would completely upset the wave action taking place in the exhaust system.

Of the joints in the exhaust system, the most important is the junction of the system to the cylinder barrel as the gas is at its highest velocity, temperature and pressure at this point. Thus the transition from barrel port to exhaust pipe must be smooth and any gasket used must not, under any circumstances, protrude into the gas stream. This point is most important and a sudden change in the power characteristics of the engine can easily be due to this. Because of this effect, a series of experiments can be quickly carried out using different size gaskets which may yield some useful information as to the suitability of the system as a whole. Naturally, once these experiments have been carried out, the system should be modified in the light of the findings and the correct gasket used in order to achieve the optimum use of the exhaust gas energy.

The practical installation of the exhaust pipe may raise a considerable number of problems when the more complex systems are employed. Often the pipe has to avoid various frame members while at the same time, be well tucked in so that it does not ground when cornering and, in the case of moto-cross machines, is not likely to suffer damage from rocks or other machines. However, the pipe must not be mounted too close to the engine or the heat radiated from the pipe will raise the temperature of the crankcase which will in turn lower the weight of charge passing into the engine and thus the power output. For this reason, an improvement in performance can often be gained with a road racing engine if the streamlining is arranged to duct air not only over the barrel, and exhaust

port in particular, but also over the exhaust system. This point has already been mentioned in the previous chapter in the notes on exhaust pipe finishes and it should not be overlooked that the more efficiently the exhaust pipe radiates its heat, the more likely is the engine to pick up this heat with consequent loss of power.

Further points to consider when fitting an exhaust system to a machine are the position of the outlet and the operation of gearchange and rear brake pedals. The outlet must point to the rear of the machine without inconveniencing following riders (standard regulations), be out of harm's way but not allow the exhaust gas to impinge on the rear wheel. If the outlet, in particular the tailpipe of the more complex systems, is not tucked in it may easily be damaged if the rider is involved in a melee at the start. With riders all jostling for position, starting machines, riding side-saddle, climbing aboard and avoiding slow starters this can easily happen and at best knock the edge off the performance and at worst completely upset the action of the system so that the engine may run weak and the piston either seize or melt its crown.

If the outlet is directing the exhaust gases either onto or very close to the rear wheel trouble may arise from one of two factors. The first is oil on the tyre due to the oil laden exhaust gases of many two-strokes. This may not be too much of a problem with a machine that is well developed and has the carburation correctly set but during the development stages the engine may easily go through a phase of blowing much of the charge straight through and out of the exhaust system. This can be sufficient to soak the rear wheel on one side in the course of a short race and is naturally far more of a problem when a pipe with megaphone is fitted than with any other type of system. If it is not appreciated at the start that this is likely to happen full realisation may dawn when the machine goes into a slide on a corner halfway through the race.

APPENDIX

SOME formulæ which may be used when calculating the size of the exhaust system.

1in. = 25·4 mm. 10 mm. = ·3937in.

$1\text{in.}^3 = 16{\cdot}387\ \text{cm.}^3$

Cubic capacity of cylinder $= \frac{\pi}{4} d^2 l$ where d = bore of cylinder, l = stroke of engine.

Capacity is in cm.^3 if d and l are in cm., and in in.^3 if d and l are in inches.

Volume of straight pipe is also $= \frac{\pi}{4} d^2 l$ where d = inside diameter of pipe, l = length of pipe.

Note that in all cases when calculating the volume of the exhaust system that the length is taken along the centre line of the pipe. Volume of a frustum of a cone $= \frac{\pi}{12} L\,(D^2 + Dd + d^2)\text{in}^3$.

i.e. a megaphone
where D = large diameter in inches
d = small diameter in inches
L = length in inches

The included angle of the megaphone may be calculated from D, d and L as follows:

$$\text{Pipe angle} = 2\ \text{Tan}^{-1} \frac{\frac{(D\text{-}d)}{2}}{L}$$

For example, to calculate the volume and cone angle of a megaphone 30in. long with diameters of 1¼in. and 3in.

$$\text{Volume} = \frac{\pi}{12}\ .30\ (3^2+(3\times 1{\cdot}25)+1{\cdot}25^2)\ \text{in.}^3$$

$$= \frac{\pi}{12}.30\ (9+3{\cdot}75+1{\cdot}56) = \frac{\pi}{12}\ .30.14.31$$

$$= \underline{112{\cdot}3\text{in.}^3}$$

$$\text{Angle} = 2\ \text{Tan}^{-1} \frac{\frac{(3\text{-}1{\cdot}25)}{2}}{30} = 2\ \text{Tan}^{-1} \frac{{\cdot}875}{30} = 2\ \text{Tan}^{-1}\ {\cdot}02917$$

$$= 2\times 1°40' = \underline{3°20'}$$

Conversely if the smaller diameter and megaphone angle are known the following may be used to find the larger diameter at any length.

$$D = d + 2L\ \text{Tan}\ \frac{\theta}{2}\ \text{where}\ \theta = \text{included angle.}$$

Thus to find the larger diameter of a megaphone 30in. long with an included angle of 4°, and a small diameter of 1·25in.

D = 1·25+2.30. Tan 2° = 1·25+60. ·0349
= 3·344″

By using the formulæ for the volumes of a pipe and a conic frustum the total volume of a resonant pipe may be found.

Thus if a pipe has dimensions as follows: port diameter 1·25in., megaphone junction diameter 3in., tailpipe diameter 1in., first and second megaphone lengths 30in. and 6in. respectively, and tailpipe length 8in.

$$\text{Volume of first megaphone} = \frac{\pi}{12}\ {\cdot}30\ (3^2+(3\times 1{\cdot}25)+1{\cdot}25^2)$$

$$= \underline{112{\cdot}3\text{in.}^3}$$

$$\text{Volume of second megaphone} = \frac{\pi}{12}\ {\cdot}6\ (3^2+3\times 1+1^2) = \frac{\pi}{4}{\cdot}2{\cdot}13$$

$$= \underline{20{\cdot}4\text{in.}^3}$$

$$\text{Volume of tailpipe} = \frac{\pi}{4}\ {\cdot}1^2{\cdot}8$$

$$= \underline{6{\cdot}3\text{in.}^3}$$

$$\text{Then volume of megaphones} = 112{\cdot}3+20{\cdot}4$$

$$= \underline{132{\cdot}7\text{in.}^3}$$

$$\text{Total volume of system} = 132{\cdot}7+6{\cdot}3$$

$$= \underline{139\text{in.}^3}$$

$$\text{Also total length of system} = 30+6+8$$

$$= \underline{44\text{in.}}$$

Dec. 14, 1937. M. KADENACY 2,102,559

EXPLOSION OR INTERNAL COMBUSTION ENGINE

Filed Aug. 1, 1934

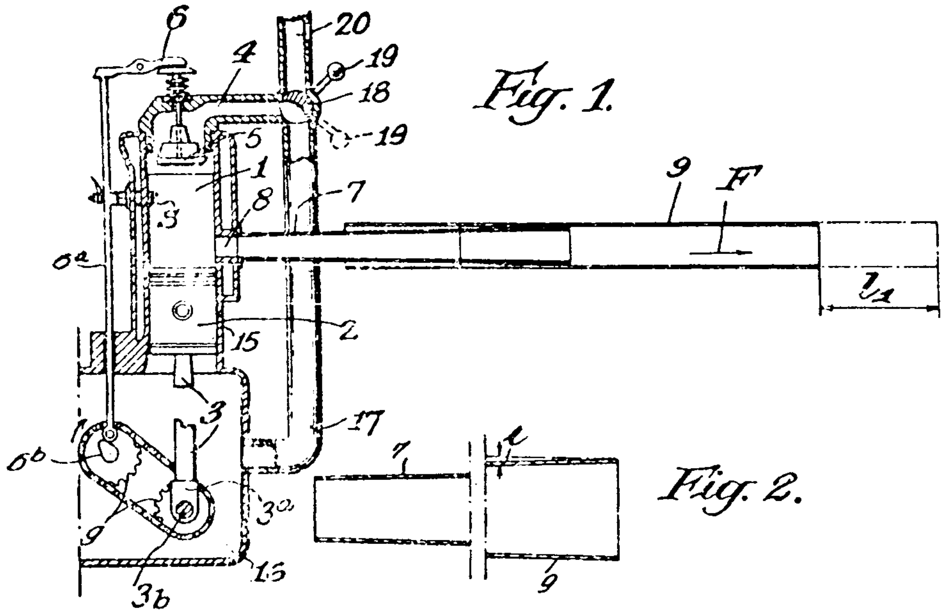


Fig. 1.

Fig. 2.

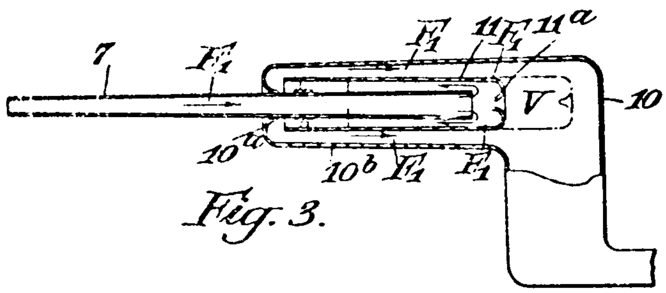


Fig. 3.

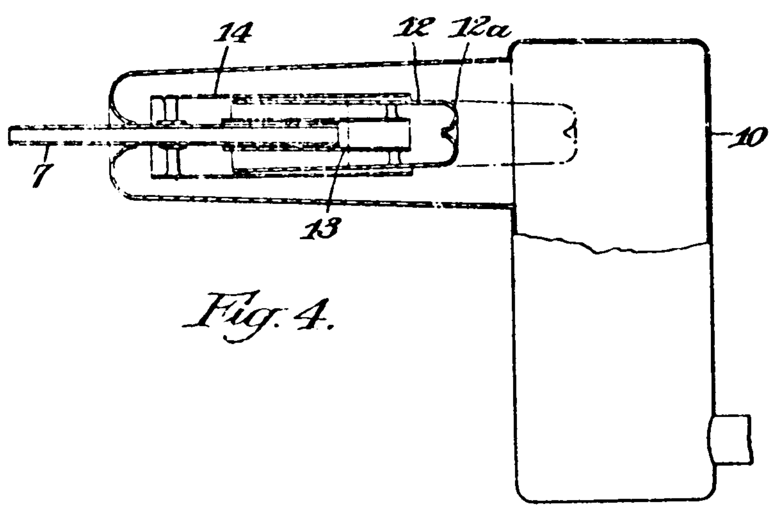


Fig. 4.

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Patented Dec. 14, 1937 2,102,559

UNITED STATES PATENT OFFICE

2,102,559

EXPLOSION OR INTERNAL COMBUSTION ENGINE

Michel Kadenacy, Paris, France

Application August 1, 1934, Serial No. 738,014
In France August 1, 1933

9 Claims. (Cl. 60—32)

The doctrine of evacuating the cylinder of a two stroke cycle internal combustion engine by permitting the burnt gases to leave the cylinder through the exhaust orifice as a mass which is governed by the laws of reflection and rebound has been established by the applicant in his prior British specifications Nos. 269,181, 308,593, 308.594 and 308,595.

In these specifications the applicant has for the first time stated that as a natural consequence of the combustion of the charge the burnt gases while still in the cylinder, form a body having the properties similar to those of a resilient body and behaving as though it possessed a high initial velocity while still in the cylinder. The applicant has stated that if this body is allowed to do so it will leave the cylinder through the exhaust orifice as a mass, which in itself is the consequence of its high initial velocity, and it will evacuate the cylinder completely or substantially completely. Subsequently, as a consequence of its impact upon the external gaseous medium and of the fact that the velocity of the said mass is greater than the possible velocity of compression of the external gaseous medium, the reaction from the latter will cause the said mass of burnt gases to rebound into the cylinder, and all the above mentioned phenomena occur when no exhaust system is provided in continuation of the exhaust orifice.

The present invention relates to such engines, that is to say to engines wherein the burnt gases are evacuated from the cylinder through the exhaust orifice as a mass or body.

In the applicant's above mentioned prior specifications he has proposed, in such an engine, to utilize the evacuation of the cylinder for the purpose of recharging the cylinder, but at the time when such specifications were filed, his researches had led him to believe that it was essential in the operation of such engines to close the exhaust orifice in such a manner that the resulting vacuous condition in the cylinder remained available for securing the entry of the subsequent charge.

The applicant has now been led by his further consideration of the phenomena in question to devise a method of charging such engines which is independent in principle of the closure of the exhaust orifice and which avoids all the practical objections that arise when use is made of the means described in the above mentioned prior British specifications.

This method consists in arranging for the inlet orifice to open, at the normal speed of the engine, after the exhaust orifice opens but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the cylinder, and cause a suction effect to be exerted in the cylinder at the said inlet orifice.

If the inlet is opened in the manner specified, air or a carburetted mixture of air and fuel according to the type of engine under consideration is suddenly drawn into the cylinder and the latter is filled satisfactorily so much the more as the gases thus admitted at a high speed are, on account of their inertia, compressed to a pressure greater than that existing in the inlet duct; and this timing of inlet establishes the engine as a two-stroke cycle engine.

The return of the tail end of the column of exhaust gases tends to occur after the cylinder has been filled with fresh gases, but the re-entry of this tail end of the exhaust column into the cylinder is resisted by the charge of fresh gases contained therein, so that the following cycle normally takes place under the most satisfactory conditions.

Now the intervals elapsing between the opening of the exhaust orifice, the mass exit of the burnt gases, and the return of this mass to the cylinder are intervals of time which are substantially independent of engine speed and consequently they extend over larger crank angles at high engine speeds than at low engine speeds.

It therefore follows that for an engine in which the valve gear, on the one hand, and the exhaust device, on the other hand, have fixed characteristics, there is also for this engine a definite working speed having a lower limit and an upper limit relatively close to each other, from which this working speed cannot depart without the engine ceasing to operate in the desired manner.

According to the invention means are provided whereby the engine is enabled to operate in the desired manner between wider working limits, as will be hereinafter described.

The invention further provides means for allowing this engine to start and to reach a working speed for which the above phenomena can occur.

These means consist either in a valve or other suitably arranged equivalent device, or again in a device for the injection, into the cylinder, of air under a relatively low pressure, upon starting. This injection of air is adapted to prevent the return of the exhaust gases into the cylinder, and to thus allow the operation until the engine has acquired sufficient speed.

Various forms of carrying out the subject-mat-

ter of the invention will be described hereinafter, with reference to the accompanying diagrammatic drawing, in which:

Figure 1 is a longitudinal section through an engine cylinder provided with an exhaust pipe.

Figure 2 is a detail of the said exhaust pipe.

Figure 3 shows a modified form of exhaust pipe.

Figure 4 shows another modified form of exhaust pipe.

By way of example, it will be assumed that this device is applied to an engine, in the cylinder (or in each cylinder) 1 of which slides a piston 2 driven by the crank 3a on the crank shaft 3b through the medium of a connecting rod 3.

4 designates an inlet conduit on the orifice of which is fitted a valve 5 controlled for instance by a push-rod 6a and rocker arm 6. The push-rod 6a is actuated by a cam 6b on the usual cam shaft, being geared by means of timing gears g—g (illustrated in the present embodiment as of 1–1 ratio) to the crank shaft 3b. The conduit 4 serves for the admission of the carburetted mixture if an explosion engine is under consideration, or of air if an internal combustion engine is considered and, in the latter case, the cylinder would also carry an injector for the admission of the fuel at the end of the compression. The present embodiment illustrates an explosion engine having a suitable ignition element, e. g., the spark plug S.

The exhaust takes place through a conduit 7 opening in the cylinder by means of a port 8 uncovered by the piston, when the latter comes in proximity to its lower dead centre, and, according to the invention, the characteristics of the distribution and exhaust devices are such that the inlet is opened at the time the gaseous column is formed, according to the method described, in the conduit 7 and in the device following it, this column escaping at a high speed towards the exterior.

The suction effect resulting therefrom in the cylinder thus causes the latter to be completely filled up with fresh air or with carburetted mixture, and prevents the subsequent return, into this cylinder of the tail end of the column of exhaust gases.

According to a particular arrangement of the invention, the exhaust device is adjustable in such a manner that the chronological law of the phenomena taking place therein can be varied at will, and that it is thus possible, as previously explained, to widen the limits between which the working speed of the engine can vary.

By way of example, the figures of the accompanying drawing illustrate devices comprising an exhaust conduit the useful length of which can be varied.

The body of gases in this conduit is subjected to corresponding variations and, consequently, it is more or less rapidly shaken by the body of gases issuing from the cylinder at the end of the expansion, and which strikes against it and rebounds several times as previously explained.

It results therefrom that by lengthening the exhaust conduit, the moment at which occurs the formation of the column producing a shock similar to a water hammer in the cylinder, is retarded, and, consequently, this new arrangement corresponds to a lower speed of the engine.

Reversely, a shortening of the exhaust conduit corresponds to a higher speed of the engine.

In the form of construction illustrated in Figure 1, the exhaust conduit comprises the fixed tube 7 previously mentioned, and, at the end of this fixed tube, a sliding tube 9.

The position illustrated in full lines corresponding to a definite working speed, in order to be able to reduce this speed, the tube 9 is moved in the direction indicated by the arrow F, up to a position such as that shown in dot and dash lines. The increase l_1 of the useful length of the exhaust conduit involves an increase of the body of gases contained in this conduit, and, consequently, causes the setting in motion of the gases, in the form of a column having a rapid movement, to be retarded.

Another particular arrangement of the invention, adapted to appreciably promote the phenomena above described, consists in that the exhaust device is arranged for facilitating the movement of the gases in the direction for exhaust, and, on the contrary, for checking it in the reverse direction.

In the example illustrated in Figure 1, and in detail and on an enlarged scale in Figure 2, the tube 7 is in the shape of a frustum having an inclination i and outwardly flared, this inclination i being of the order 1 to 2 per cent., for instance.

The increase of the section of the passageway for the gases in proportion as they escape, facilitates this movement, but, reversely, if the tail end of the gas column expands backwardly, this section decreases and hinders said expansion; this concurs to prevent the return of the gases into the cylinder and, consequently, to further improve the operation.

In the form of construction shown in Figure 3, the fixed tube 7 extends in the exhaust box 10, and its end is covered within this exhaust box by a sliding tube 11, the bottom 11a of which is closed, so that the gas column escapes according to the path indicated by the arrows F_1. The bottom 11a of the sliding tube 11, and the portion 10a of the wall of the exhaust box 10 through which passes the tube 7, constitute deflectors facilitating the changes in the direction of the movement of the gases.

The structure is so arranged that the gas column escaping at high speed in the direction F, enters the exhaust box after it has produced its entire useful effect, that is, when it has reached substantially the limit of its outward travel from the cylinder.

By moving the sliding tube 11 to a position such as that shown in dot and dash lines, the volume of gases contained in the exhaust tube is increased according to a quantity V, and this new arrangement corresponds to a lower speed of the engine.

Finally, according to an arrangement similar to that described, the tube 7 and the tubular portion 10b of the exhaust box 10 which surrounds the tubes 7 and 11 have outwardly flared shapes.

According to the form of construction illustrated in Figure 4, the tubular portion 11 of the preceding device is replaced by another tube 12, having also a closed bottom 12a, and rigid with a guide 13 sliding on the tube 7, and, moreover, another tube 14 slides on the tube 7 and on the end of the tube 12, so that by moving both these tubes 12, 14, simultaneously or separately, a greater range of adjustment is available, and this adjustment can be effected more accurately.

It is to be understood that, in order to actuate the adjusting means described, or any other equivalent means, any suitable control can be provided.

Finally, for allowing the starting and speeding up of the engine, that is to say, its operation at abnormal working speeds, for which the phenomena described no longer take place in the same conditions, and would no longer ensure satisfactory working of the engine, the invention provides the use of any suitable device arranged for blowing air into the cylinder and thus preventing the return of the exhaust gases. The air pressure necessary is moreover very small, and the device to be used can be of the most simple type.

For example, as shown in Figure 1 the inlet duct 4 may communicate through a two-way rotary valve 18, having a handle 19, with a duct 20 open to the atmosphere and a duct 17 opening into the crank case 16, the cylinder comprising a third port 15 for the admission of air into the crankcase in the well known manner.

I claim:

1. Method of controlling two-stroke cycle internal combustion engines, which comprises establishing communication between the cylinder and exhaust system during the firing stroke, maintaining permanent open communication between the cylinder and the atmosphere while the exhaust port is open, providing for the issuance of the burnt gases from the cylinder substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gases moves outward and thereafter returns toward the cylinder, preventing the entrance of fresh charging air until the said issuance of the burnt gases is in full progress, admitting fresh charging air into the cylinder when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, while the exhaust port is still open, and providing for the said fresh charge to occupy the cylinder in the interval elapsing between the said exit of the burnt gases and the instant when the pressure of the returning gases becomes effective within the cylinder, whereby the re-entry of the said burnt gases into the cylinder will be opposed by the fresh charge contained therein.

2. Method of controlling two-stroke cycle internal combustion engines, which comprises establishing communication between the cylinder and exhaust system during the firing stroke, providing for the issuance of the burnt gases from the cylinder substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gases moves outward and thereafter returns from a point which may be within the exhaust system, providing a permanent free passage for the burnt gases to the limit of outward travel of said burnt gases, preventing the entrance of fresh charging air until the said issuance of the burnt gases is in full progress, admitting fresh charging air into the cylinder when he said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, while the exhaust port is still open, and providing for the said fresh charge to occupy the cylinder in the interval elapsing between the said exit of the burnt gases and the instant when the pressure of the returning gases becomes effective within the cylinder, whereby the re-entry of the said burnt gases into the cylinder will be opposed by the fresh charge contained therein.

3. Method of controlling two-stroke cycle internal combustion engines, which comprises establishing communication between the cylinder and exhaust system during the firing stroke, admitting fresh charging air into the cylinder under pressure when starting and running up to normal speeds, providing at normal running speed for the issuance of the burnt gases from the cylinder substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gases moves outward and thereafter returns from a point which may be within the exhaust system, providing a permanent free passage for the burnt gases to the limit of outward travel of said gases, preventing at normal running speed the entrance of fresh charging air until the said issuance of the burnt gases is in full progress, admitting at normal running speed fresh charging air into the cylinder, when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, while the exhaust port is still open, and providing for the said fresh charge to occupy the cylinder in the interval elapsing between the said exit of the burnt gases and the instant when the pressure of the returning gases becomes effective within the cylinder, whereby the re-entry of the said burnt gases into the cylinder will be opposed by the fresh charge contained therein.

4. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the exhaust conduit providing a permanent free passage for the burnt gases to the limit of outward travel of said gases, and providing a passage for the gases during their outward motion as a mass having no cross section of substantially greater area than any cross section thereof further from the cylinder.

5. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust duct on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, an exhaust conduit in continuation of the said exhaust orifice, the said conduit providing a permanent free passage for the burnt gases to the limit of outward travel of said gases and providing a passage for the gases during their outward motion having a cross section increasing progressively away from the cylinder.

6. A two-stroke cycle internal combustion engine as claimed in claim 5, the said passage pro-

vided by the exhaust conduit having a taper of the order of 1 to 2 per cent.

7. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the said conduit providing a permanent free passage for the burnt gases to the limit of outward movement of said gases and providing a passage for the gases during their outward motion as a mass having no cross section of substantially greater area than any cross section thereof further from the cylinder and no abrupt and substantial increase in volume for a length substantially equal to the limit of outward travel of the mass of burnt gases.

8. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the said conduit providing a permanent free passage for the burnt gases to the limit of outward travel of said gases and providing a passage for the gases during their outward motion as a mass having no cross section of substantially greater area than any cross section thereof further from the cylinder, a silencer in continuation of the said conduit, the said silencer being situated at such a distance along said pipe that the said gases substantially terminate their outward motion before entering said silencer.

9. The device as claimed in claim 4 in which the exhaust conduit comprises telescoping tubes.

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March 15, 1938. M. KADENACY 2,110,986

EXHAUST DEVICE FOR EXPLOSION OR INTERNAL COMBUSTION ENGINES

Filed Aug. 1, 1934

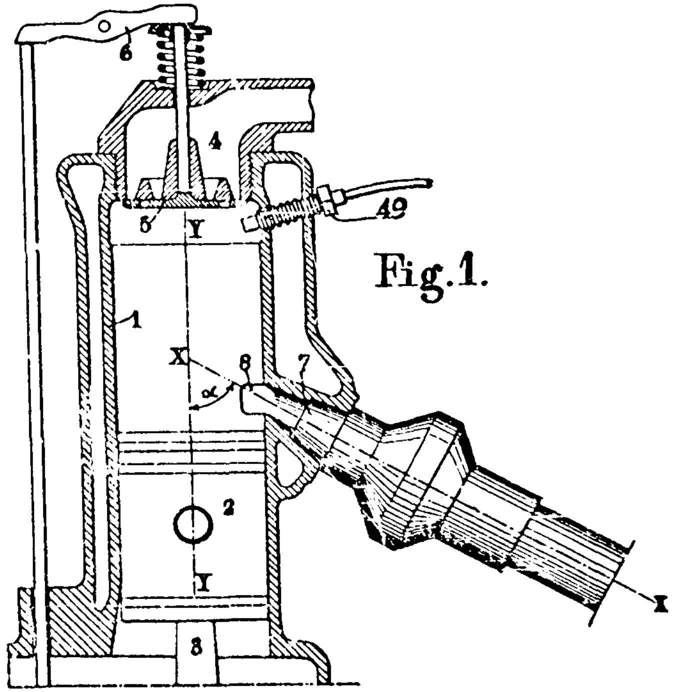


Fig.1.

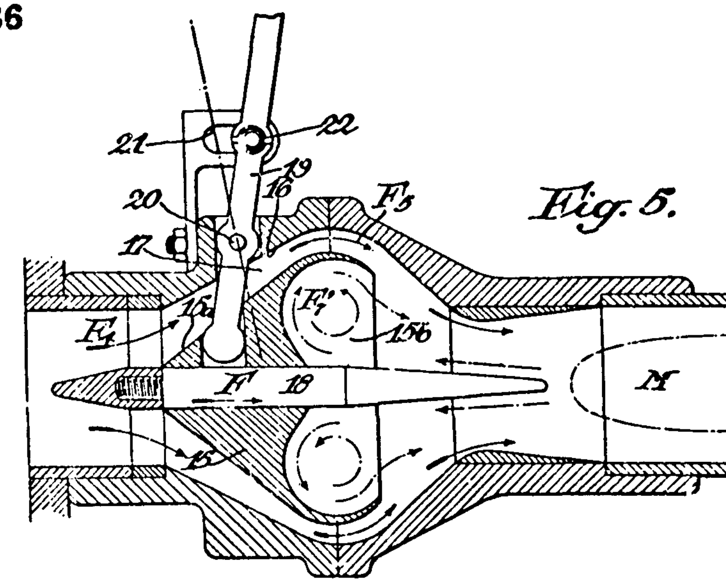


Fig.5.

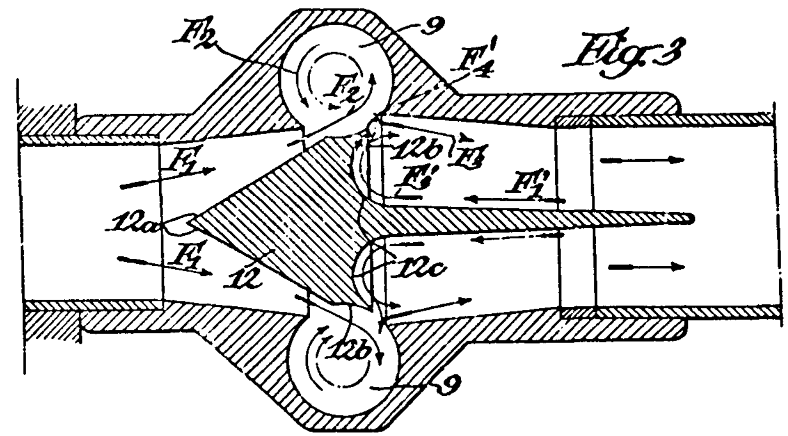
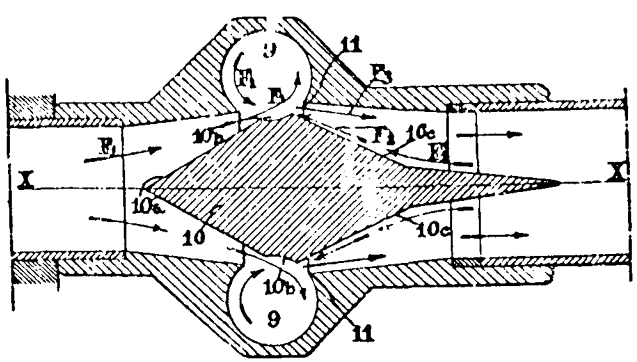


Fig.3

Fig.2.



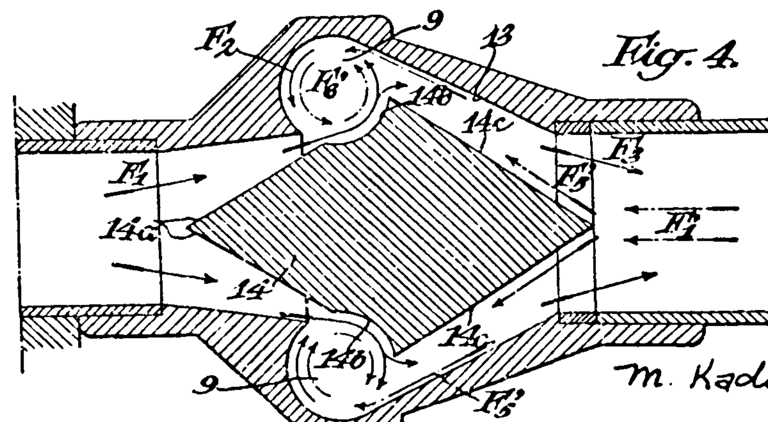


Fig.4.

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Fig. 6.

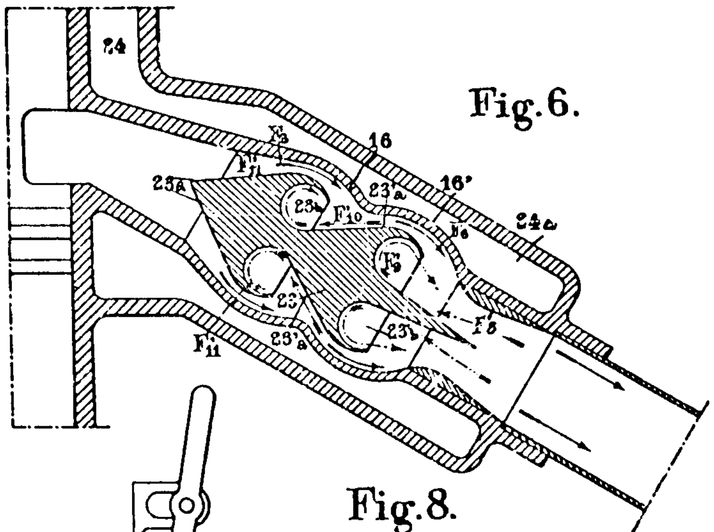


Fig. 8.

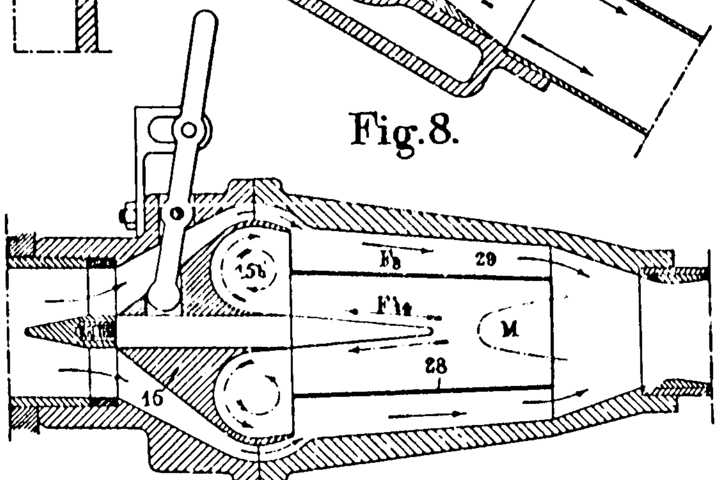


Fig. 7.

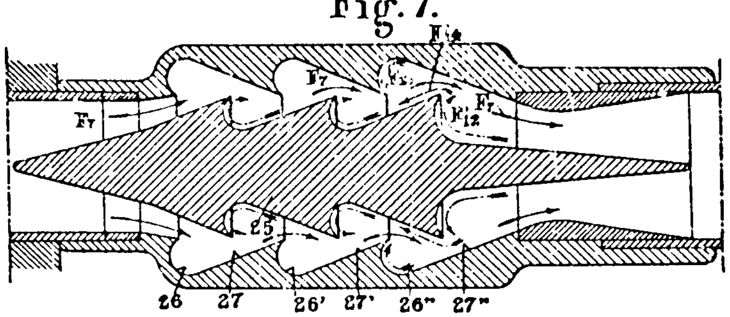


Fig. 9.

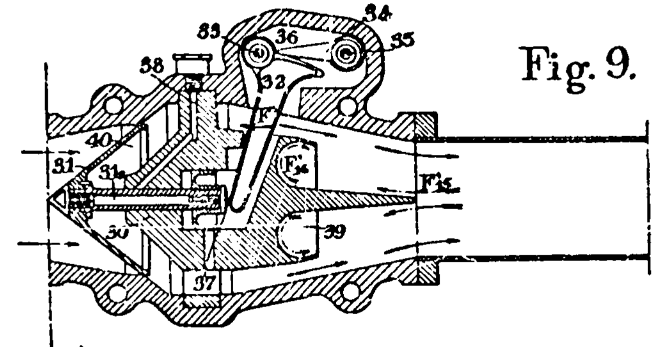


Fig. 10.

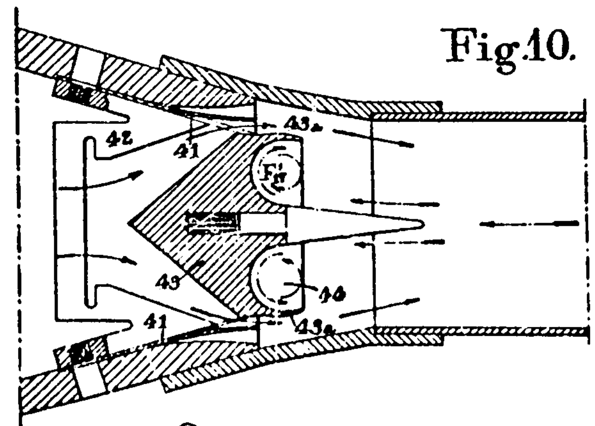
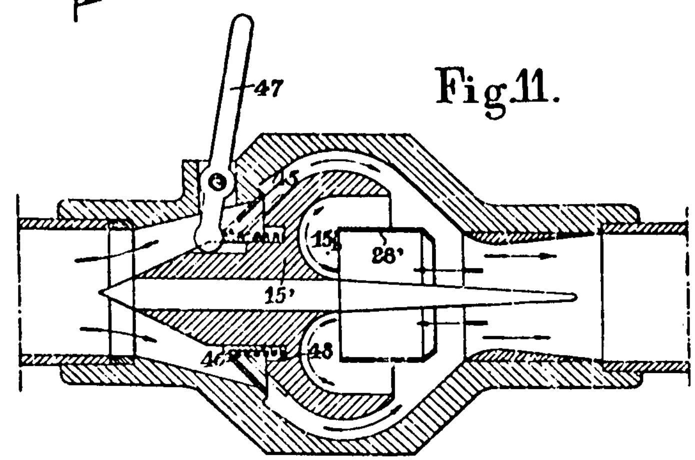


Fig. 11.



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Patented Mar. 15, 1938 2,110,986

UNITED STATES PATENT OFFICE

2,110,986

EXHAUST DEVICE FOR EXPLOSION OR INTERNAL COMBUSTION ENGINES

Michel Kadenacy, Paris, France

Application August 1, 1934, Serial No. 733,016
In France August 1, 1933

7 Claims (Cl. 60—32)

The applicant has found that in an internal combustion engine, the behaviour of the gases is such as to lead to the conclusion that as a consequence of the combustion of the charge, and while still in the cylinder, the burnt gases form a mass having a high initial velocity and possessing properties similar to those of a resilient body, so that when the exhaust orifice opens this mass seeks to project itself bodily from the cylinder and to leave the latter in a consequent vacuous condition.

The present invention relates to two-stroke cycle internal combustion engines, wherein at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when an adiabatic flow only is involved and in such a short interval of time that it is discharged as a mass leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder.

The applicant has also found that in the operation of such an internal combustion engine the burnt gases do not leave the cylinder immediately the exhaust orifice commences to open. There is first a period of delay, during which the burnt gases do not issue from the cylinder and after this delay has elapsed the burnt gases issue bodily from the cylinder with an extremely high velocity as a mass which responds to the laws of reflection and rebound and it leaves in the cylinder a profound depression. Subsequently, this outward motion of the burnt gases is reversed in direction and if the gases are allowed to re-enter the cylinder they destroy the depression left therein.

In Patent Number 2,102,559 dated December 14, 1937, the applicant has described and claimed a method of charging two-stroke cycle internal combustion engines which consists in opening the inlet orifice for the introduction of a fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust system as a consequence of their mass exit from the cylinder and cause a suction effect to be exerted in the cylinder at the said inlet orifice.

In such an engine, an untimely return of the burnt gases may have an objectionable influence on the contents of the cylinder and the object of the present invention is to provide in the exhaust system of an engine operating in accordance with the above method, means which are adapted to permit the free exit of the mass of burnt gases from the cylinder and to prevent the return of this mass or a portion thereof of burnt gases to the cylinder by utilizing the above mentioned properties of the burnt gases.

The invention consists in the provision within the exhaust system of deflecting means situated between the exhaust orifice and the point in the exhaust system from which the return of the burnt gases occurs after the said bodily exit from the cylinder at high velocity, the said deflecting means being adapted to permit the free outflow of the burnt gases and to prevent by reflection the return of the said gases to the cylinder.

It is to be understood that in considering the free outflow of the burnt gases losses due to friction are excepted, but these losses should of course be kept down to a minimum.

Now if the characteristics of the exhaust system are fixed, the interval elapsing between the bodily exit of the burnt gases from the cylinder and the subsequent reversal in direction of movement of these gases is a duration of time which is substantially constant. As a consequence, this interval will extend over a larger crank angle at high engine speeds than at low engine speeds. Consequently the return of the burnt gases is more liable to exert an objectionable influence on the contents of the cylinder at low engine speeds. If the angular separation between exhaust opening and inlet opening is also fixed, there will be a limiting low speed for which the return occurs too soon to permit the timing of inlet opening to be operative in the required manner.

In general, therefore, the means according to the invention will have the effect of improving the operation of the engine at low engine speeds and of extending the possible range of working speeds of the engine in the direction of low speeds.

Preferably the said deflecting surfaces will be situated close to the cylinder since the point where the body of gases issuing from the cylinder rebounds on the external gaseous medium is itself situated very near the exhaust orifice of the cylinder, and it is indispensable to place the device according to the invention between this exhaust orifice and said rebounding point in order that the obturation should be effected according to the method described.

The invention further provides an arrangement applicable in particular to engines for which very low speeds must be obtained relatively to their normal working speed.

In fact, in this case, the quantity of exhaust gases and their speed have very low values relatively to those corresponding to normal working conditions.

It is then difficult to form and maintain an obturating plug in the conditions above set forth, and this so much the more as, if the distribution gear is adjusted for producing, between the opening of the exhaust and that of the inlet for the following admission, a retardation corresponding to a definite fraction of the cycle, the duration of this retardation is reversely proportional to the speed of the engine; it results therefrom that the plug, already less important and less stable, as just stated, should however be able to "hold" a longer time for allowing the charging of the cylinder to take place.

A first means provided by the invention for avoiding this inconvenience consists in the combination, with the above walls, of a valve or other equivalent device, the operation of which is satisfactory at very low working speeds.

Moreover, even if, at the beginning of the exhaust, this valve does not close with the desired rapidity, the formation of the plug by the body of gases gives time for this closing to take place, whereupon, even if the plug is prematurely destroyed, the valve remains closed and maintains the obturation during a sufficient time.

The invention further provides another means which consists in suitably modifying the distribution gear, and particularly, the lapse of time between the exhaust and admission, in order that the period of time during which the obturating plug must "hold" should be, at low speed, reduced in the required proportions.

Such a device can be constructed in any suitable manner, for instance, by means of a sleeve or other adjustable obturator co-operating with the distribution ports, of a mechanism allowing to modify the angular position of the cams or other parts controlling the valve gear, etc.

This device can be controlled either by hand, or automatically, for instance, by the variations of speed of the engine, or by the operation of the device controlling the admission of fuel (throttle valve of the carburettor, injection pump) etc.

Use can also be made, for running in the exceptional working conditions obtained during starting and until the engine has acquired sufficient speed, of the means described in the above mentioned patent application, and which consists in blowing air into the cylinder.

In the accompanying drawings, Figures 1 to 11 illustrate various forms of carrying out the subject-matter of the invention, which will be described hereinafter by way of example only.

In these drawings:

Figure 1 is a diagrammatic view of an engine cylinder, provided with an exhaust device according to the invention.

Figure 2 is a sectional view of the exhaust device shown in Figure 1.

Figure 3 shows a modified form of exhaust device.

Figure 4 shows another modified form of exhaust device.

Figure 5 shows a further modified form of exhaust device provided with adjusting means.

Figure 6 shows a further modified form of exhaust device provided with cooling means.

Figure 7 shows a further modified form of exhaust device provided with adjusting means.

Figure 8 shows an exhaust device of multiple form.

Figure 9 shows a form of exhaust device combined with a non-return valve for use at low engine speeds.

Figure 10 shows a similar arrangement to Figure 9 with a different form of nonreturn valve.

Figure 11 shows another form of exhaust device in combination with a nonreturn valve and means for putting the said valve out of action at normal running speeds.

In these figures, the arrows indicate the displacement of the gases during their to-and-fro or whirling movement at a high speed utilized for the production of the obturating plug; the arrows in full lines correspond to the forward movement, and the arrows in dot and dash lines correspond to the return movement after rebounding.

By way of example, it is assumed that this device is applied to an engine having a cylinder 1, in which slides a piston 2 actuating the crank shaft through the medium of a connecting rod 3. 4 designates the inlet conduit, and 5 the inlet valve controlled for instance by a push-rod and rocker arm 6.

7 designates the exhaust conduit opening in the cylinder through one or more ports 8 uncovered by the piston 2 when it comes near its lower dead centre.

This engine can be of the explosion type in which case it will be fed through the conduit 4 with carburetted mixture, or of the combustion type, in which case it will be fed with air only through the conduit 4 and with fuel through an injector 49.

According to the invention, on the exhaust conduit 7 are arranged walls adapted to trap and guide, in the conditions set forth, the mass formed by the exhaust gases, whilst it has a movement at high speed, before it returns to the cylinder 1.

A first feature of this device resides in the fact that the axis X X of the exhaust conduit is preferably set downwardly, that is to say it forms, with the axis Y Y of the cylinder, a relatively acute angle α. More generally, the exhaust conduit is set relatively to the cylinder, in order to impart a change of direction as small as possible to the body of gases, that is to say to check the same to the least possible extent.

According to Figures 1, 2, the inner wall of the conduit 7 forms a toroidal cavity 9, opposite which is arranged a deflector 10 according to the axis X X.

This deflector has a conical portion 10a, the apex of which faces the cylinder 1, and said deflector is so set as to guide the gases, when they issue from the cylinder, towards the torus 9, according to F_1.

The gases, trapped and guided by the inner wall of the torus, as well as by a portion 10b of the deflector 10 which completes this torus, whirl according to F_2 and form an obturating plug.

When the whirling movement comes to an end, the gases issue to the exterior through a free space 11 existing between the deflector 10 and the edge of the torus 9.

A portion of the gases can also pass directly through this space 11 without being trapped by the torus 9, and, according to the path indicated at F_3, rebounds as above set forth and returns according to F'_1. It then encounters the deflector 10, the down side end of which is formed, as indicated at 10c, for sending it into the torus 9 according to F'_2.

This fraction of the gases then encounters the plug formed at F_2 and is prevented from returning to the cylinder.

By the time the whirling movement of the plug has come to an end, all the gases contained in the

conduit form a column which rapidly escapes, as stated.

Figure 3 illustrates a modification comprising, in the wall of the exhaust conduit, a torus 9 similar to that of the form of construction previously described, and opposite this torus, a deflector 12 constituting two walls 12a, 12b, which also guide the gases according to F_1—F_2.

The difference relatively to the preceding form of construction resides in the fact that the down side end of the deflector is shaped, at 12c, as a portion of a torus, so as to trap the gases which may have passed the torus 9 according to F_3 and have rebounded according to F'_1, and to cause them to form a second plug F'_3, or again to send them, according to F'_4, tangentially to the first plug so that they reinforce the same instead of running the risk of destroying it, as in the first embodiment in which they are radially admitted.

In the modification illustrated in Figure 4 the torus 9 formed by the inner wall of the exhaust conduit, terminates, on the down side, in a conical incline 13.

The deflector 14 has, on the up side, two parts 14a—14b which deflect the gases in the direction F_1—F_2, so as to form a whirling plug as described. On the down side, it has a part 14c parallel, or nearly parallel, to the wall 13.

In these conditions, the fraction of the gases which has escaped according to F_3 and has rebounded according to F'_1 is guided, between both walls 14c—13, according to F'_5, so as to tangentially enter the torus 9 and to form an eddy F'_6 which is added to the eddy F_2 and reinforces the obturating plug.

In the preceding embodiments, the obturating plug is formed by the totality or a portion of the gases during their forward movement. Devices will be described hereinafter in which the walls according to the invention are arranged for allowing the entire body of gases to freely pass during its forward movement, and to trap it and form the obturating plug during its return movement.

The form of construction illustrated in Figure 5 comprises a deflector 15, arranged in a bulged portion 16 of the exhaust conduit and so shaped as to present to the gases, on the up side, a smooth and continuous surface 15a of conical shape, terminating in a convex portion, in the shape of a fraction of a torus, and, on the down side, a cavity or recess 15b also in the shape of a torus.

When they issue from the cylinder, the gases enter the space 17 existing between the walls 15a and 16, and, these walls having a smooth and continuous shape, the movement and the peculiar properties of the gases are not subjected to any perturbation. These gases escape at a high speed according to F_4—F_5, they rebound on the atmosphere, and return, also at a high speed, in the form of a resilient body such as indicated at M. This body M is trapped by the recess 15b in which it forms an obturating plug by whirling according to F'_7.

The following arrangement has also been provided in this apparatus:—

When the working speed of the engine increases, it is advantageous to provide for the gases issuing from the cylinder a passageway of larger cross section, in order to avoid checking these gases, the volume of which, and probably also the speed, are more important. Reversely, when the working speed diminishes, it is advantageous to always maintain the same conditions of operation by reducing the section of the passageway, and by adjusting the walls of the device in such a manner that they always trap the body of gases, upon its return, in the same conditions.

For that purpose, in the embodiment shown in Figure 5, the deflector 15 is slidably mounted on a support 18 arranged according to the axis of the exhaust conduit, in such a manner that it can be moved in the direction of the arrow F for diminishing the section of the passageway 17 and moving the recess 15b towards the mass or body M when the working speed of the engine diminishes, or reversely, when this working speed increases.

This sliding movement is controlled from the exterior by a lever 19 pivoted about a fixed stud 20; this lever can itself be actuated by hand, and, eventually, it can be held stationary in any position, for instance by means of a slotted frame 21 and of a clamping screw 22.

Figure 6 illustrates a form of construction of the same type as the preceding one, in which the deflector 23 comprises two elements arranged in series and each forming, on the up side, a smooth and continuous wall 23a, 23'a for allowing the gases to freely escape during the forward movement according to F_6 and, on the down side, a recess 23b, 23'b in the shape of a torus.

Both these elements are respectively located opposite two bulged portions 16, 16' of the exhaust conduit.

The body of gases, when it returns towards the cylinder according to F'_8, is trapped by the down side recess 23'b and forms, by whirling according to F'_9, an obturating plug; if a portion of this return wave passes beyond the recess 23'b, according to F'_{10}, it is trapped by the up side recess 23b which compels it to form, by whirling according to F'_{11}, another plug, so that it is possible, by means of these stepped recesses, to prevent in all cases any return of the gases to the cylinder.

It is easy, if need be, to cool the walls of this device, in any suitable manner according to Figure 6, for instance, the water jacket 24 for cooling the cylinder extends, at 24a, about the exhaust conduit.

Use can of course be made, according to circumstances, of deflectors having any number of elements in series. According to Figure 7, for instance, the deflector 25 comprises three of these elements.

Moreover, the inner wall also forms, towards the down side, cavities or recesses adapted to trap the gases which may have passed beyond those of the deflector.

These recesses are indicated in the drawings, at 26, 26', 26'', respectively opposite each of the elements of the deflector, and they terminate in smooth and continuous walls 27, 27', 27'' adapted to allow the gases to freely flow on their forward movement, according to F_7.

Upon their return movement, the gases are trapped in particular by the down side recess of the deflector 25, according to F'_{12}; another portion is trapped by the down side recess 26'' of the exhaust conduit, according to F'_{13}; the portion of the return wave which has succeeded in passing beyond both these recesses is trapped by the following recesses of the deflector and exhaust conduit.

This combination of both series of deflectors provides, under a very reduced volume, six cavities or recesses ensuring a very high efficiency of the device.

Figure 8 illustrates a device of the same type as

that shown in Figure 5, and in which the following improvement has been made; on the down side of the deflector 15, a sleeve 28 is arranged within the exhaust conduit and is concentric with the latter.

Upon their forward movement, the gases are guided according to F_1 in the annular space 25 existing between this sleeve and the exhaust conduit. Upon its return movement, the body of gases M enters this sleeve and is guided by the same, according to F'_{14}, towards the cavity or recess 15b which thus traps it in better conditions.

Another advantage of this device consists in that, when a fraction only of the body of gases has rebounded on the external atmosphere, whilst the other fraction has not yet rebounded, both these fractions are separated by the sleeve 28, and, instead of rubbing against each other they respectively rub on the outer and inner smooth walls of this sleeve.

The checking effect exerted on the gas is thus considerably reduced, and this so much the more as the speed of each of the two fractions of gases relatively to the sleeve is half the speed they have relative to each other.

Figures 9 to 11 illustrate devices comprising the combination of deflectors adapted to trap the bodies of gases and of obturating valves, which combination is mainly applicable to engines which are to reach very low speeds, as previously explained.

According to Figure 9, a support 30 is mounted in the exhaust conduit and is perforated, towards the up side, with an axial channel in which slides the tail 31a of an automatic obturating valve 31.

This valve 31 is urged towards its closing position by a lever 32 pivotally mounted on a fixed stud 33, and urged in opposite direction to the arrow F' by a tappet 34 pivotally mounted on a stud 35 and controlled by any suitable returning spring.

The studs 33, 35, as well as the returning spring, are arranged in a casing 36 placed outside the exhaust conduit.

The end of the lever 32 bears on an adjustable push piece 37 constituting the end of the valve tail piece 31a.

The lubrication of this valve tail piece is ensured by a lubricator 38.

On the other hand, the support 30 forms, on the down side, a cavity or recess 39 which traps the return wave, returning according to F'_{15}, and guides it so as to form a plug according to F'_{16}.

This obturation, ensured by the combined action of the recess 39 and valve 31, can be further improved by the use of a valve so arranged as to constitute a wall which traps and suitably guides the body of gases. In the example under consideration, the valve 31 is in the shape of a cone flared towards the down side, and constitutes a cavity or recess 40 which can trap and stop the body of gases, even if the valve is not yet reclosed. It is moreover to be noted that the action of the body of gases on the bottom of this recess 40 tends, on the other hand, to accelerate the closing movement of the valve.

According to Figure 10, the valve is constituted by very light resilient blades 41, clamped, at one end, between the inner wall of the exhaust conduit and a fixed support 42.

In closed position the free ends of these blades press upon a bearing portion 43a of an axial deflector 43.

This deflector 43 forms, on the other hand, on the down side, a recess 44, in the shape of a torus, adapted to trap and guide, according to F'_{17}, the body of gases during its return movement.

Finally, Figure 11 illustrates, in combination with a deflector 15' and a sleeve 28', of the same type as those shown in Figure 8, a check valve 45 longitudinally sliding on the deflector 15' and urged towards its closing position by a returning spring 46.

This valve 45 in combination with the toric recess 15'b formed on the down stream side of the deflector 15' ensures the obturation of the exhaust duct at low working speeds.

At normal working speed, the obturation is ensured solely by the body of gases suitably trapped and guided, the valve 45 being maintained out of action, in the position indicated in dot and dash lines, by a push piece 47.

In this position, the returning spring 46 is enclosed in a closed recess 48 and is not subjected to the action of the gases.

In the above examples, one cylinder only has been particularly considered, but any suitable arrangements can of course be provided for rendering the invention applicable to a multi-cylinder engine.

I claim:

1. In an internal combustion engine having a cylinder, a piston moving in this cylinder and connected to a crank shaft and inlet and exhaust orifices on the cylinder for introducing a combustible mixture and discharging the products of combustion respectively, and wherein for recharging the cylinder the exhaust gases are allowed to leave the cylinder substantially in their entirety and the fresh charge is allowed to enter the cylinder in the interval occurring between the above-mentioned exit of the burnt gases and the subsequent return movement of the burnt gases, an exhaust conduit leading from said exhaust orifice, a body within said exhaust conduit at a zone located nearer the exhaust orifice than the point of return of the burnt gases, and means for supporting said body concentrically with respect to said exhaust conduit, said body having a toroidal cavity directed away from the engine cylinder and having a progressively decreasing configuration in the direction of said cylinder, whereby a free outflow is provided for the burnt gases but their return to the cylinder is hindered by the formation in and by said toroidal cavity of a whirling gaseous plug consisting of at least a portion of the returning burnt gases.

2. In an internal combustion engine having a cylinder, a piston moving in this cylinder and connected to a crank shaft and inlet and exhaust orifices on the cylinder for introducing a combustible mixture and discharging the products of combustion respectively, and wherein for recharging the cylinder the exhaust gases are allowed to leave the cylinder substantially in their entirety and the fresh charge is allowed to enter the cylinder in the interval occurring between the above-mentioned exit of the burnt gases and the subsequent return movement of the burnt gases, an exhaust conduit leading from said exhaust orifice, a body located within said exhaust conduit in spaced relation with respect to the inner wall thereof and at a zone nearer the exhaust orifice than the point of return of the burnt gases, and means for supporting said body coaxially with respect to said exhaust conduit, said body having the form of a cone with its apex directed towards the cylinder and having a toroidal cavity in its base whereby a free outflow is provided for the burnt gases but their return to the

cylinder is hindered by the formation in and by said toroidal cavity of a whirling gaseous plug consisting of at least a portion of the returning burnt gases.

3. In an internal combustion engine having a cylinder, a piston moving in this cylinder and connected to a crank shaft and inlet and exhaust orifices on the cylinder for introducing a combustible mixture and discharging the products of combustion respectively, and wherein for recharging the cylinder the exhaust gases are allowed to leave the cylinder substantially in their entirety and the fresh charge is allowed to enter the cylinder in the interval occurring between the above-mentioned exit of the burnt gases and the subsequent return movement of the burnt gases, an exhaust conduit leading from said exhaust orifice, said exhaust conduit having at a zone located nearer the exhaust orifice than the point of return of the burnt gases, an intermediate portion of greater internal cross-sectional area than the cross-sectional area at the cylinder, a body located within said intermediate portion, and means for supporting said body coaxially with respect to said exhaust conduit, said body having the form of a cone with its apex directed towards the cylinder and having a toroidal cavity in its base whereby a free outflow is provided for the burnt gases but their return to the cylinder is hindered by the formation in and by said toroidal cavity of a whirling gaseous plug consisting of at least a portion of the returning burnt gases.

4. In an internal combustion engine having a cylinder, a piston moving in this cylinder and connected to a crank shaft and inlet and exhaust orifices on the cylinder for introducing a combustible mixture and discharging the products of combustion respectively, and wherein for recharging the cylinder the exhaust gases are allowed to leave the cylinder substantially in their entirety and the fresh charge is allowed to enter the cylinder in the interval occurring between the above-mentioned exit of the burnt gases and the subsequent return movement of the burnt gases, an exhaust conduit leading from said exhaust orifice, a body located within said exhaust conduit in spaced relation with respect to the inner wall thereof and at a zone nearer the exhaust orifice than the point of return of the burnt gases, means for supporting said body coaxially with respect to said exhaust conduit with freedom for axial movement, said body having the form of a cone with its apex directed towards the cylinder and having a toroidal cavity in its base whereby a free outflow is provided for the burnt gases but their return to the cylinder is hindered by the formation in and by said toroidal cavity of a whirling gaseous plug consisting of at least a portion of the returning burnt gases and means for axially displacing said body within said exhaust conduit whereby the section of passage between said body and the inner wall of the exhaust conduit may be varied.

5. In an internal combustion engine having a cylinder, a piston moving in this cylinder and connected to a crank shaft and inlet and exhaust orifices on the cylinder for introducing a combustible mixture and discharging the products of combustion respectively, and wherein for recharging the cylinder the exhaust gases are allowed to leave the cylinder substantially in their entirety and the fresh charge is allowed to enter the cylinder in the interval occurring between the above-mentioned exit of the burnt gases and the subsequent return movement of the burnt gases, an exhaust conduit leading from said exhaust orifice, a body within said exhaust conduit at a zone located nearer the exhaust orifice than the point of return of the burnt gases, means for supporting said body concentrically with respect to said exhaust conduit, said body having a toroidal cavity directed away from the engine cylinder and having a progressively decreasing configuration in the direction of said cylinder, whereby a free outflow is provided for the burnt gases but their return to the cylinder is hindered by the formation in and by said toroidal cavity of a whirling gaseous plug consisting of at least a portion of the returning burnt gases, a sleeve located within said exhaust conduit immediately beyond said body and means for supporting said sleeve concentrically with respect to said exhaust conduit whereby the burnt gases during their outflow pass between the exhaust conduit and the sleeve and on their return pass within said sleeve to said toroidal cavity.

6. In an internal combustion engine having a cylinder, a piston moving in this cylinder and connected to a crank shaft and inlet and exhaust orifices on the cylinder for introducing a combustible mixture and discharging the products of combustion respectively, and wherein for recharging the cylinder the exhaust gases are allowed to leave the cylinder substantially in their entirety and the fresh charge is allowed to enter the cylinder in the interval occurring between the above-mentioned exit of the burnt gases and the subsequent return movement of the burnt gases, an exhaust conduit leading from said exhaust orifice, a body within said exhaust conduit at a zone located nearer the exhaust orifice than the point of return of the burnt gases, means for supporting said body concentrically with respect to said exhaust conduit, said body having a toroidal cavity directed away from the engine cylinder and having a progressively decreasing configuration in the direction of said cylinder, whereby a free outflow is provided for the burnt gases but their return to the cylinder is hindered by the formation in and by said toroidal cavity of a whirling gaseous plug consisting of at least a portion of the returning burnt gases and an automatic valve located in said exhaust conduit nearer the cylinder than said toroidal cavity and adapted to open and allow the burnt gases to pass beyond said body and to reclose and prevent the return of the burnt gases to the engine cylinder.

7. The combination as claimed in claim 6 including means for maintaining said automatic valve open except at low engine speeds.

MICHEL KADENACY.

April 5, 1938. M. KADENACY 2,113,480

DISTRIBUTION SYSTEM FOR EXPLOSION OR INTERNAL COMBUSTION ENGINES

Filed Aug. 1, 1934

Fig. 1.

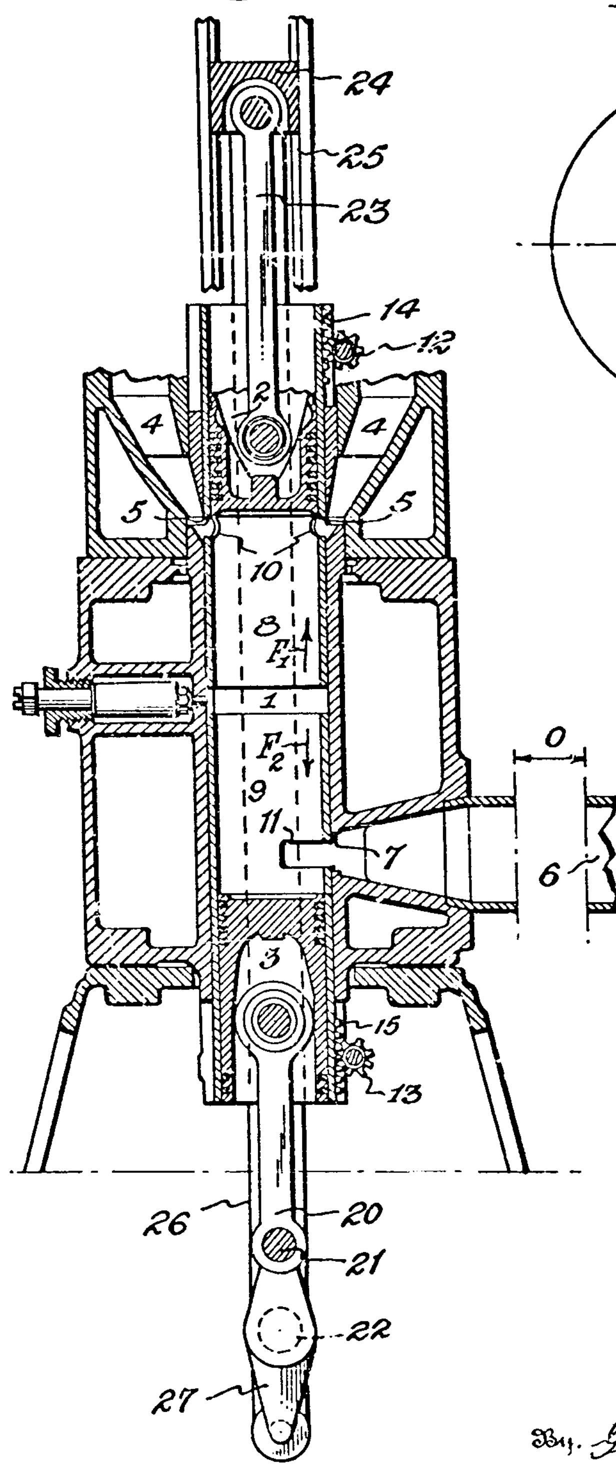
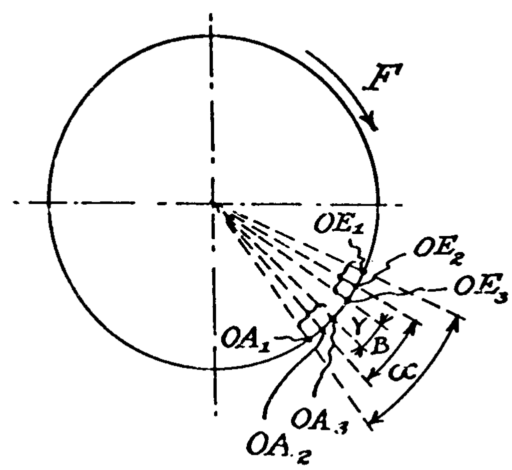


Fig. 2.



Inventor.

M. Kadenacy

By Glascock Downing & Seebold Att'ys.

Patented Apr. 5, 1938

2,113,480

UNITED STATES PATENT OFFICE

2,113,480

DISTRIBUTION SYSTEM FOR EXPLOSION OR INTERNAL COMBUSTION ENGINES

Michel Kadenacy, Paris, France

Application August 1, 1934, Serial No. 738,015
In France August 1, 1933

1 Claim. (Cl. 60—32)

The present invention relates to two stroke cycle internal combustion engines wherein the vacuum or high depression left in the cylinder by the discharge of the burnt gases substantially as a mass in an interval of time shorter than that required for the burnt gases to expand down to the ambient atmospheric pressure by adiabatic flow is utilized for introducing the fresh charge through an inlet which is opened while the exhaust port is open and when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, all as set forth in the applicant's Patent No. 2,102,559, granted December 14, 1937.

The object of the invention is to provide an improvement in the operation of such engines whereby the output may be maintained over a wide range of speeds.

In the accompanying drawing, by way of example:

Figure 1 is a diagrammatic view of an engine according to the invention.

Figure 2 is a diagram showing the action of the device for controlling the distribution.

It will be assumed, for instance, that an engine, in the cylinder (or in each cylinder) 1 of which slide two diametrally opposed pistons 2, 3, is under consideration. The piston 3 is connected by a rod 20 with the crank arm 21 of a crank shaft 22. The other piston 2 is connected by a rod 23 with a cross head 24 slidably mounted in guides 25. The cross head 24 is connected by a rod 26 with a crank arm 27 carried by the crank shaft 22 and arranged in opposed relation to the crank arm 21.

Admission takes place through two conduits 4 opening in the wall of the cylinder 1 through ports 5, and exhaust is effected through a conduit 6 opening in the wall of the cylinder 1 through a port 7. Both ports 5 and 7 are respectively uncovered by the pistons 2 and 3, when the latter approach their "bottom" dead centre, that is to say the position opposed to that corresponding to maximum compression.

This engine operates in the following manner.

At the end of the firing stroke, the piston 3 uncovers the port 7, and, the body of burnt gases resiliently rebounds between the cylinder wall and the gaseous atmosphere existing in the conduit 6, whereupon, under the influence of the repeated shocks to which it has been subjected, this gaseous atmosphere is in its turn set in motion, and the burnt gases leave the cylinder as a mass forming with the gaseous atmosphere, in the conduit 6, a column which escapes at a high speed and produces in the cylinder a shock similar to a negative water hammer, that is to say a suction effect.

It is at this moment that the piston 2 uncovers the ports 5, so that the cylinder is automatically and completely filled up, and, when the gas column, checked on the down side by the atmosphere it encounters, tends to return into the cylinder, its re-entry into the latter is opposed by the charge of fresh air (or of carburetted mixture) which has just been admitted therein.

According to the invention, the above condition is maintained at varying engine speeds by the provision of an adjusting device whereby the lapse of time between the opening of the exhaust and that of the inlet can be varied.

In the example under consideration, this adjustment is obtained by means of sliding sleeves 8, 9, interposed between the cylinder 1 and the pistons 2 and 3, respectively, and provided with ports 10, 11 opposite the ports 5, 7.

The arrangement and the respective levels of the various ports are such that by causing the sleeves 8, 9 to slide in the cylinder, the distribution is modified; by moving the sleeve 8 in the direction of F_1, the opening of the inlet is retarded, and, by moving the sleeve 9 in the direction of F_2, the opening of the exhaust is retarded, the reverse effects being obtained if the sleeves 8 and 9 are moved in the opposite directions to those indicated.

These displacements are controlled by pinions 12, 13 respectively meshing with racks 14, 15, longitudinally arranged on the outer walls of the sleeves 8, 9.

Figure 2 is a circular diagram showing various positions of the distribution system; F designates the direction of rotation of the engine; the points OE1, OE2, OE3 designate different positions of exhaust opening, and the points OA1, OA2, OA3, different positions of inlet opening.

When the positions of the sleeves 8, 9 are such that the exhaust opens at OE1, and the admission at OA1, the angular retardation of the admission relatively to the exhaust has a relatively great value α. This arrangement corresponds to a high speed of rotation of the engine, since, notwithstanding this high angular value, this retardation must have, in terms of time, a value which is always relatively small in order that the charging shall be effected before the tail end of the column of exhaust gases has had the time to return into the cylinder.

If the sleeve 9 is moved in the direction of the arrow F_2, the opening of the exhaust port is re-

tarded, the exhaust then takes place at a point such as OE2, and, if the sleeve 8 is moved in a direction opposed to that of the arrow F_1, the opening of the inlet is advanced, the admission then occuring at a point such as OA2.

For this new arrangement, the angular retardation of the opening of the inlet relatively to that of the exhaust is now only β and this, from what has been stated, corresponds to a lower speed of the engine.

Finally, the arrangement EO3—OA3 of Figure 2 diagrammatically illustrates a still smaller angular retardation δ corresponding to a still lower speed of the engine.

It is of course to be understood that the engines to which the present invention is applied may comprise any of the expedients described in Patent No. 2,102,559 above referred to.

I claim:

A two stroke cycle internal combustion engine having a cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, the said conduit providing a permanent free passage for the burnt gases to the limit of outward travel of said gases, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, and means for varying the angular retardation of admission relative to exhaust, whereby the desired timing of inlet opening may be established for varying engine speeds.

MICHEL KADENACY.

July 25, 1939. M. KADENACY 2,167,303

EXHAUST DEVICE FOR INTERNAL COMBUSTION ENGINES

Filed Aug. 31, 1935

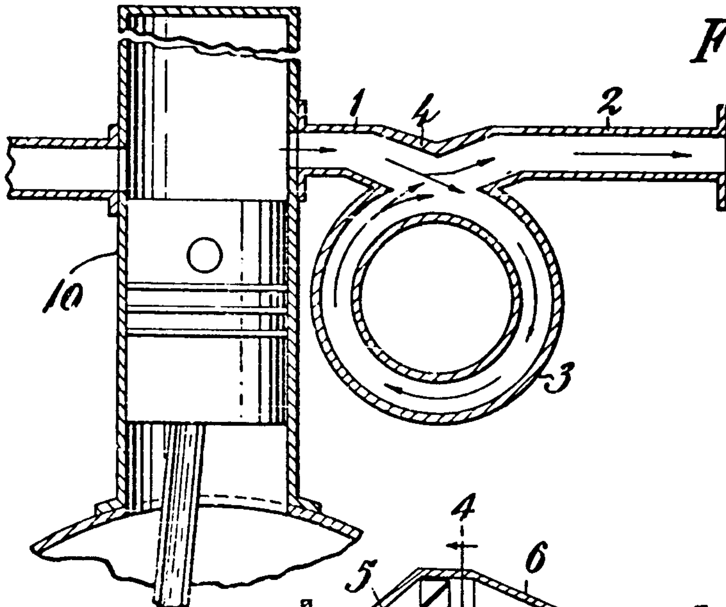


Fig.1

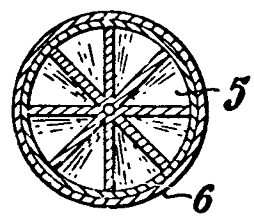


Fig.4

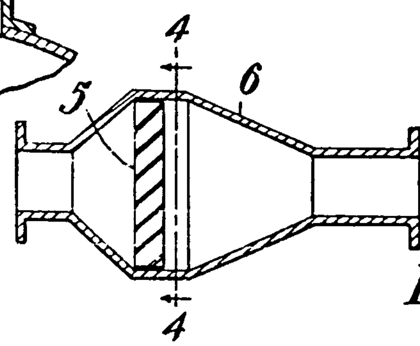


Fig.2

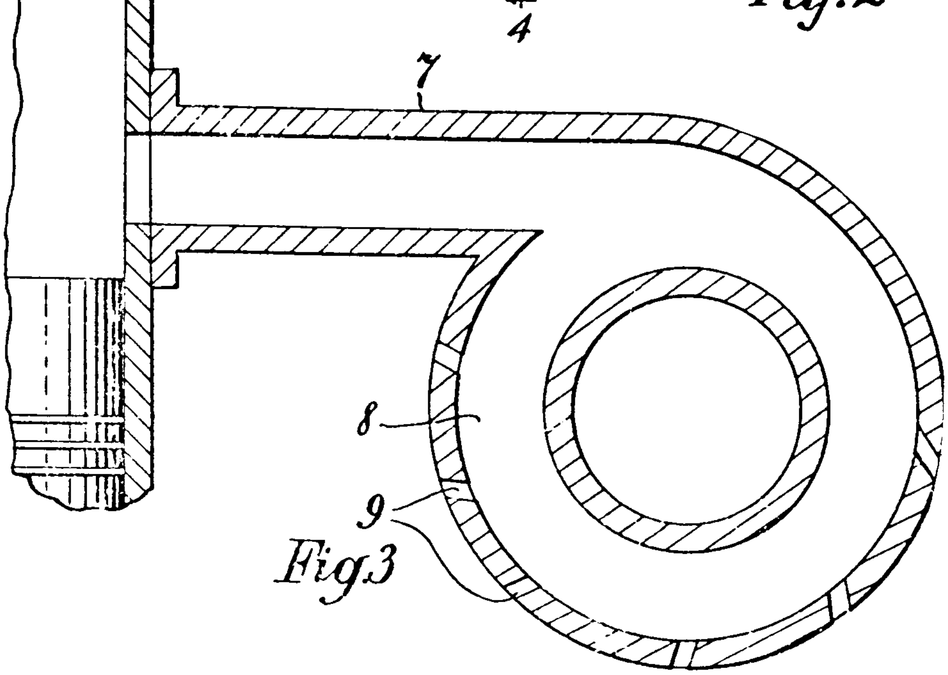


Fig.3

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Patented July 25, 1939 2,167,303

UNITED STATES PATENT OFFICE

2,167,303

EXHAUST DEVICE FOR INTERNAL COMBUSTION ENGINES

Michel Kadenacy, Paris, France

Application August 31, 1935, Serial No. 38,826
In Great Britain August 31, 1934

2 Claims. (Cl. 60—32)

The invention relates to two-stroke internal combustion engines, wherein the evacuation of the cylinder by the bodily displacement of the burnt gases from the cylinder into the exhaust system is utilised for the purpose of introducing a fresh charge by controlling the inlet orifice to open for the introduction of the said charge when the exhaust gases are moving outwardly through the exhaust port or duct as a consequence of this bodily displacement from the cylinder.

In such an engine two disturbing factors occur which may have an objectionable action upon the charge introduced into the cylinder.

When the gases have reached the end of their outward travel as a consequence of their bodily displacement from the cylinder their movement is reversed in direction and the return impact, if the exhaust port is then still open, may cause the charge in the cylinder to be fouled and some of this charge to be forced out of the inlet ports.

In any case when this reversal in direction occurs it will tend to arrest the further entry of air through the inlet ports.

Further, if the inlet ports close at the end of the charging period before the exhaust ports are closed and the mass of burnt gases is then still moving outwardly through the exhaust system, this may cause some of the charge contained in the cylinder to be drawn into the exhaust system through the open exhaust ports.

The object of the invention is to provide means whereby the objectionable actions of an untimely return of the burnt gases or of a prolonged suction in the exhaust system may be minimised.

The invention will now be described with reference to the accompanying drawing, in which:—

Figure 1 relates to one embodiment of the invention in which the issuing mass of burnt gases is directed on to a small gaseous mass in the exhaust system and exerts its pressure on the main atmosphere in a continuous manner.

Figure 2 relates to an embodiment in which the explosion gases are constrained to follow an elongated spiral path.

Figure 3 relates to an embodiment in which the explosion gases escape to the atmosphere gradually.

Figure 4 is a section of Figure 3 on line 4—4 looking in the direction of the arrows.

In the arrangement illustrated in Figure 1, the exhaust pipe comprises a short duct 1 for connection to the cylinder 10, a duct 2 in continuation of the duct 1 and leading to the atmosphere, and between these two portions an annular chamber 3 arranged as a by-pass on the main exhaust pipe 1, 2 and in open communication with both 1 and 2.

The body of explosion gases leaving the engine cylinder encounters a reflecting surface 4 on the wall of the duct opposite the entry to the chamber 3, which is situated so as to direct the said body into the annular chamber 3, the volume of which will be substantially the same as that of the engine cylinder.

The gases then follow by a series of reflections a continuous path in the chamber 3 while exerting a component of their force upon the main mass in the duct 2.

The face of the gaseous column in the duct 2 in this case forms a reflecting surface on which the gases impinge, and which yields as the atmospheric column in the duct 2 is put into movement in the direction of exhaust.

In the arrangement illustrated in Figure 2, an assembly of fixed guide vanes or blades 5 is mounted in the exhaust duct 6 in order to impart to the issuing explosion gases a whirling or screw motion.

This assembly will be fitted as close as possible to the cylinder and between the engine and the silencer. The area of passage for the gases will be made such that the volume of gases dealt with can pass freely.

The blades 5 act as deflecting surfaces and, in combination with the enclosing walls of the duct 6, impose rotary forward movement on the explosion gases whereby the absolute velocity of the latter is not appreciably reduced but the relative speed of forward motion is reduced but never wholly destroyed.

The body of explosion gases is thus made to act like a screw action upon the atmospheric mass in the duct 6, so as to exert a continuous pressure on the latter, until this atmospheric resistance is broken down and the explosion gases can escape.

The arrangement shown in Figure 3 differs from the foregoing in that the exhaust duct is no longer, properly speaking, freely open to the atmosphere.

In this case the exhaust duct opens into the annular chamber 8, forming a closed circuit, situated preferably as close as possible to the latter and having a volume a little larger than the cylinder volume.

The explosion gases project from the cylinder and encounter deflecting surfaces formed by the walls of the duct 7 which direct them into the chamber 8 and in this chamber they undergo a

series of deflections which compel them to follow a continuous closed path in this chamber.

The peripheral walls of the chamber 8 are provided with small orifices 9 leading to the atmosphere and suitably directed to facilitate the escape of the explosion gases.

The explosion gases circulating without loss of absolute velocity in the chamber 8 are thus projected gradually into the external atmosphere through the orifices 9 and any return to the cylinder by rebounding from the external atmospheric mass is thus prevented.

It will be understood from the foregoing that all the devices described are adapted to cause the issuing mass of burnt gases to travel along an elongated path without thereby increasing the resistance to motion of the gases along this path and that the devices do not obstruct the outflow of the burnt gases in any way or comprise any surfaces which would tend to reflect the mass of burnt gases back towards the cylinder.

It will also be understood from the foregoing that in order to be effective, the devices according to the invention must be placed nearer the cylinder than the point from which the rebound of the gases to the cylinder would occur, as they will otherwise have no influence in preventing this rebound.

With all the devices hereinbefore described, the vacuum is maintained in the cylinder during a period of time amply sufficient to permit the cylinder to be recharged by atmospheric pressure if desired.

The objectionable effect of a return of the burnt gases towards the cylinder is attenuated or wholly destroyed. A loss of charge due to a prolonged suction in the exhaust system is also reduced or avoided completely according to the position in which the device is placed in the exhaust system, since it will be understood that the volume of the void left in the cylinder and in the exhaust system by the issuing mass of burnt gases will depend upon the distance the device according to the invention is placed from the cylinder.

I claim:

1. A two stroke cycle internal combustion engine wherein the burnt gases are discharged from the cylinder into an exhaust conduit substantially as a mass whereby the said mass moves outward and thereafter returns towards the cylinder from a point which may be within the said exhaust conduit, and wherein an inlet is opened for the introduction of fresh charge while the exhaust port is still open and when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, the said exhaust conduit providing a free passage for the burnt gases to the limit of outward travel of said mass and having a chamber in connection with said passage and wherein substantially the whole mass of burnt gases after leaving the cylinder is guided and is caused to adopt a whirling motion as a compact mass whereby the outward motion of the said mass is prolonged and the rebound of said mass towards the cylinder is delayed and reduced in intensity.

2. A two-stroke cycle internal combustion engine wherein the burnt gases are discharged from the cylinder into an exhaust conduit substantially as a mass whereby the said mass moves outward and thereafter returns towards the cylinder from a point which may be within the said exhaust conduit, and wherein an inlet is opened for the introduction of fresh charge while the exhaust port is still open and when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, the said exhaust conduit providing a free passage for the burnt gases to the limit of outward travel of said mass and having a chamber in connection with said passage into which substantially the whole mass of burnt gases is guided after leaving the cylinder, the axis of the chamber being in line with that of the exhaust conduit, said chamber having substantially the same volume as the engine cylinder and having a widened portion in which stationary guiding blades are positioned, which give a spiral motion to the burnt gases, whereby the outward motion of the said mass is prolonged and the rebound of said mass towards the cylinder is delayed and reduced in intensity.

MICHEL KADENACY.

Oct. 4, 1938.　　M. KADENACY　　2,131,957

CHARGING AND SCAVENGING OF THE CYLINDERS OF
TWO-STROKE INTERNAL COMBUSTION ENGINES
Filed Oct. 25, 1935

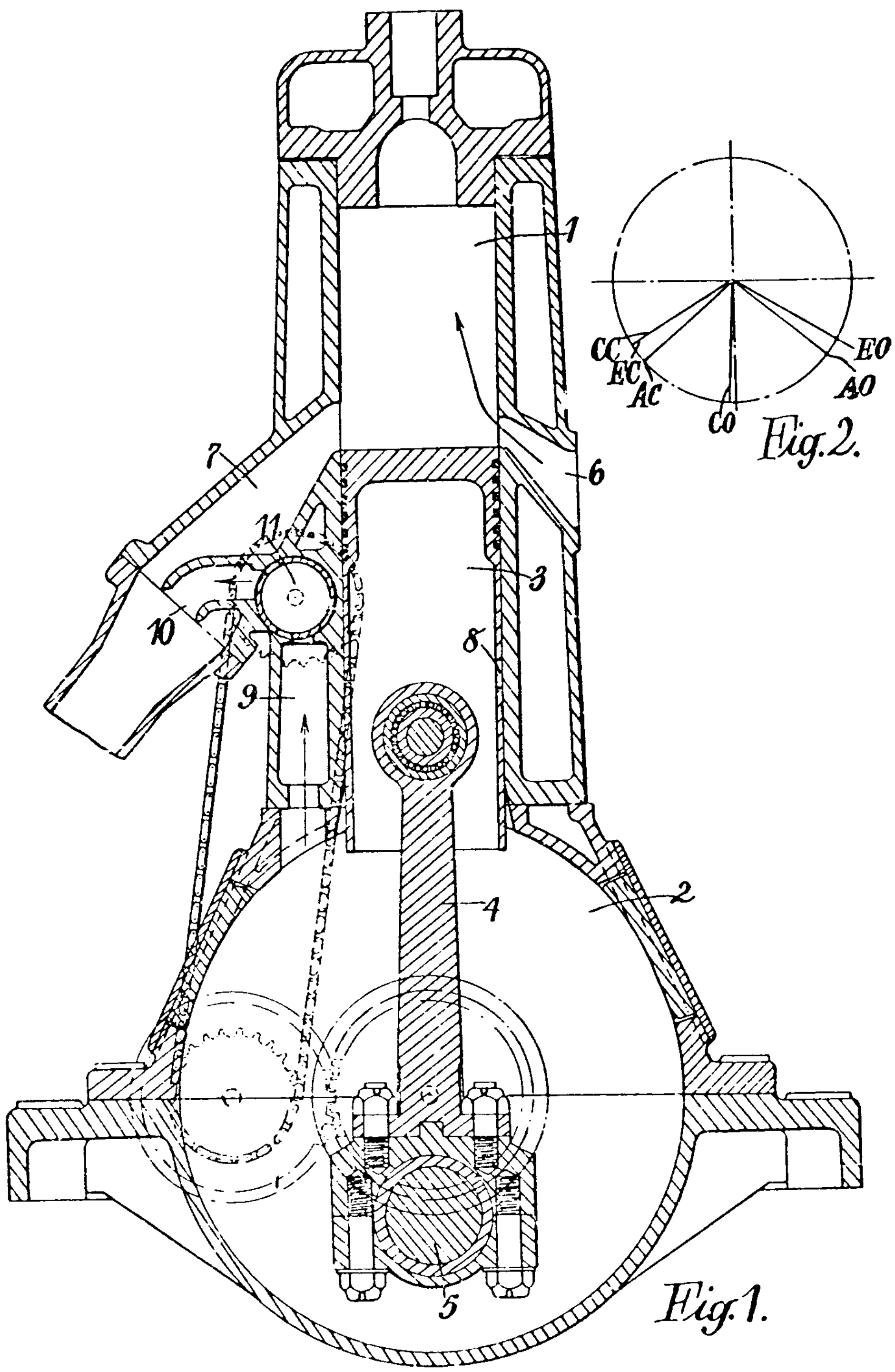

Fig.2.

Fig.1.

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Patented Oct. 4, 1938 2,131,957

UNITED STATES PATENT OFFICE

2,131,957

CHARGING AND SCAVENGING OF THE CYLINDERS OF TWO-STROKE INTERNAL COMBUSTION ENGINES

Michel Kadenacy, Paris, France

Application October 25, 1935, Serial No. 46,804
In Great Britain November 3, 1934

10 Claims. (Cl. 123—65)

This invention relates to two-stroke cycle internal combustion engines wherein the burnt gases leave the cylinder at a speed higher than that obtaining when adiabatic action only is involved and in such a short time interval that they are discharged wholly or substantially wholly from the cylinder into the exhaust system as a mass leading behind them a high depression which may reach a complete vacuum.

In such an internal combustion engine when the exhaust port opens there is first a period of delay during which no appreciable movement takes place in the gaseous medium external to the exhaust orifice and after this delay has elapsed the burnt gases issue from the cylinder as a mass at high velocity and leave a profound depression behind them in the cylinder and also in the exhaust pipe close to the cylinder. A reversal in direction of motion of the burnt gases then follows, whereupon the depression is first destroyed and then the pressure rises above atmospheric pressure in the exhaust duct adjacent the cylinder and in the cylinder itself, if the exhaust orifice is then open.

The occurrence of the above mentioned depression, its duration and its cessation can be determined by known means, and if an inlet orifice is opened in the cylinder after the exhaust orifice opens, but with only the required delay to ensure that the mass of burnt gases is then moving outwardly through the exhaust orifice or duct and causes a suction to be exerted at the inlet orifice as a consequence of said mass exit from the cylinder, a fresh charge may be admitted through the said inlet orifice by atmospheric pressure only.

Now the interval elapsing between the commencement of opening of exhaust and the reversal in direction of motion of the burnt gases subsequent to their mass exit from the cylinder is a duration of time which is largely independent of the engine speed; consequently this interval will extend over a larger crank angle at high engine speeds than at low engine speeds.

If the reversal in direction of motion of the burnt gases occurs too soon and while the exhaust orifice is still open, burnt gases may re-enter the cylinder, and if the inlet orifice is then also open some of the fresh charge may be forced out of the cylinder. On the other hand, if the inlet orifice closes while a depression still exists in the exhaust duct and the exhaust port is then still open, this may cause some of the fresh charge to be drawn out of the cylinder into the exhaust system.

The former of these objectionable actions is most likely to be exerted at low speeds and the latter at high speeds. According to the present invention a quantity of fresh gases is admitted or introduced into the exhaust system during the exhaust period. These fresh gases may serve for delaying the return of the burnt gases and obstructing their re-entry into the cylinder, or for combatting any loss of charge due to a prolonged suction in the exhaust duct, and in this way the charging of the cylinder will be improved.

In order to satisfy the requirement of the invention the fresh gases must be introduced or admitted into the exhaust system while the exhaust orifice is still open and at a point in the exhaust duct situated nearer the cylinder than the point in this duct from which the return of the burnt gases occurs, and the introduction or admission must occur after the mass exit of the burnt gases from the cylinder and before their return into the latter.

If the fresh gases are admitted by atmospheric pressure, then this admission can only be made while there is still a depression in the exhaust duct, and the means employed for this purpose must be such that they establish communication between the interior of the exhaust duct and the source of fresh gases only when this depression exists in the exhaust pipe.

If the introduction is effected under a pressure higher than atmospheric, there will normally be no advantage in commencing this introduction before the mass exit of the burnt gases occurs as this will simply increase the pressure required for effecting the introduction; and there will be no advantage in continuing it after the closure of the exhaust orifice as it will then cease to be of use.

In order to obtain an advantage from this introduction under pressure an economy must be effected in the quantity and pressure of the fresh gases injected. The injection pressure will vary according to the depression in the exhaust duct and according to the effect it is desired to obtain; and it will also vary according to the output and type of engine.

If this introduction is made with the object of combatting or delaying the return of the burnt gases, it will thus not be advantageous to effect the injection of fresh gases into the exhaust duct at moments when this high depression exists in the working chamber and in the exhaust duct.

The most advantageous moment for effecting this injection will then be immediately before the said return wave occurs, so as to oppose and even prevent the returning burnt gases from reentering the engine cylinder and pushing out the new charge admitted to the latter.

The quantity and volume of these injected gases, in order to avoid any waste of energy must, in general, be introduced not sooner than the moment when the return wave occurs and must not continue later than the closure of the exhaust orifice.

In this way the duration of the suction exerted by the cylinder will be prolonged, thereby improving the charging of the engine.

If the injection or admission of fresh gases into

the exhaust duct is effected at bottom dead centre, this will generally be found satisfactory for the purposes above stated. Most usually it will commence shortly after bottom dead centre and it is obvious that it is useless to continue this injection or admission after the closure of the exhaust orifice.

The introduction or admission of fresh gases may be so disposed relative to the closure of exhaust and the occurrence of the return of the burnt gases that the latter may overcome the inertia of the fresh gases and may even force them into the cylinder. The effect of this will be to force fresh gases into the cylinder and not burnt gases.

In the case when the exhaust orifice remains open longer than the inlet orifice, this effect may cause the final charge contained in the working chamber to be slightly above atmospheric pressure at the moment of closure of the last orifice and before the compression stage.

Further the quantity of fresh gases introduced into the exhaust duct and thereafter forced into the cylinder by the return expansion of the burnt gases may even be sufficient to form the working charge of the engine, in which case separate inlet orifices on the cylinder may be dispensed with.

One embodiment of the invention in which a charge is introduced into the exhaust pipe of an engine under pressure is illustrated in the accompanying drawing, in which—

Figure 1 is a cross section through an engine provided with means for carrying the inventon into effect.

Figure 2 is a timing diagram applicable to such an engine.

The drawing shows an engine comprising a cylinder block 1 mounted on a crank case 2, and in which moves a piston 3 driven by the connecting rod 4 from the crank shaft 5.

The cylinder is provided with piston operated inlet and exhaust ports 6 and 7 and in the example chosen, the inlet ports 6 communicate with the atmosphere and open very shortly after the exhaust ports open; the interval between exhaust opening and inlet opening being established so as to ensure that inlet opens with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust port or duct and cause a suction to be exerted at the said inlet.

The skirt of the piston comprises a further port 8 co-operating with the ports 6 when the piston rises in order to permit a charge of air to be drawn into the crank case.

The crank case also communicates through a duct 9 in the cylinder block with an injector nozzle 10 extending within the exhaust duct in the direction of outlet and situated close to the exhaust port. This duct 9 is controlled by suitable means such as the rotary valve 11 suitably actuated so as to open and close at required moments during the cycle of operations of the engine.

On the firing stroke the piston 3 opens the exhaust port 7 and then the inlet port 6 for the admission of the working charge, while compressing the air which has been drawn into the crank case 2 during the preceding upward stroke of the piston 3.

Shortly after bottom dead center, the rotary valve 11 opens and puts the crank case into communication with the nozzle 10 so that a compressed charge of air from the crank case is injected through this nozzle into the exhaust duct.

The period during which the crank case charge is transferred through the nozzle 10 into the exhaust duct is shown in Figure 2, in which EO and EC represent exhaust opening and closure, AO and AC represent the opening and closing of the inlet on the cylinder through which the fresh charge is introduced into the latter and CO and CC represent the commencement and termination of the injection into the exhaust duct. It will be seen that this injection commences shortly after bottom dead centre and terminates at or about the closure of the exhaust port.

As already mentioned the injection or admission of fresh gases into the exhaust duct may be so applied that these gases serve to charge the cylinder through the exhaust duct.

For example, the exhaust duct may comprise orifices situated close to the cylinder and controlled by automatic one way valves adapted to open into the exhaust duct when a high depression exists therein as a consequence of the mass exit of the burnt gases from the cylinder, and to close when this depression is destroyed.

Such valves may, for example, be formed of very light blade springs controlling orifices opening to the atmosphere.

If these valves or the like and the orifices they control are suitably designed, the high depression left in the exhaust duct when the burnt gases are discharged as a mass from the cylinder, will cause the valves to open and admit air into the exhaust duct. When the return occurs, these valves will automatically re-close and the fresh air enclosed in the exhaust duct will be driven into the cylinder by the return wave and thereby charge the latter through the exhaust port. In this case a separate admission port may become unnecessary.

It should be clearly understood that the fresh gases admitted or introduced into the exhaust duct may be utilized simply for improving the charging of the engine by opposing the re-entry of the burnt gases into the working chamber or additionally for providing all or a part of the fresh charge supplied to the cylinder, whether the charge is admitted by atmospheric pressure or injected under compression.

It should be noted that the action which creates the suction effect occurring in engines according to the present invention originates in the cylinder and is propagated from the cylinder into the exhaust duct, in that this suction effect is caused by the exit of at least a substantial portion of the burnt gases from the cylinder at a speed greatly in excess of the speed obtaining when adiabatic action only is involved and in such a short interval of time that it is discharged as a mass. In carrying out the present invention the natural tendency of the burnt gases to project themselves from the cylinder as a mass should be facilitated and not opposed, that is to say the area of the exhaust orifice available for the discharge of the burnt gases should be as large as possible and the interval of time in which the area required for this discharge of the burnt gases is made available should be as short as possible in order to obtain the most satisfactory results.

I claim:

1. In a two-stroke cycle internal combustion engine of the kind described the combination with a cylinder having an inlet port and an exhaust port, an exhaust duct leading from the exhaust port and means for opening inlet after exhaust opens but only with the required delay to ensure that the burnt gases are then moving

outwardly through the exhaust port and cause a suction to be exerted at inlet as a consequence of their mass exit from the cylinder, of means for introducing a quantity of fresh gases, into the exhaust duct under pressure during the exhaust period of the engine, said introduction commencing immediately before a return wave of the exhaust gases occurs.

2. In a two-stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port and an exhaust duct leading from said exhaust port, of an orifice on said duct intermediate its length, a source of compressed gaseous fluid, distribution means connecting said source of fluid to said orifice and means for controlling said distribution, means for introducing a compressed charge of gaseous fluid into the exhaust duct commencing at or about bottom dead centre and terminating not later than the closure of the exhaust port of the engine.

3. In a two-stroke cycle internal combustion engine, a cylinder, a piston or pistons in said cylinder, inlet and exhaust ports on said cylinder, an exhaust duct leading from said exhaust port, means which may comprise the said piston or pistons, for controlling said ports, said inlet port being opened after the exhaust port opens but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust port or duct as a consequence of their mass exit from the cylinder, a source of compressed gaseous fluid, an orifice in the exhaust duct close to the engine cylinder and communicating with a nozzle or nozzles extending within the exhaust duct and with said source of fluid and means for controlling communication between the said source and the said nozzle or nozzles in such a manner that the said communication will be opened during the exhaust period of the engine after the exhaust gases have issued from the cylinder and before these gases return to the cylinder.

4. In a two-stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port, and an exhaust duct leading from said exhaust port, of an orifice on said duct intermediate its length, a tubular element within said exhaust duct communicating with said orifice and extending in the direction of exhaust, a source of compressed gaseous fluid, distribution means connecting said source of fluid to said orifice and tubular element, and means for controlling said distribution means for introducing a working charge of gaseous fluid into the exhaust duct in the direction of exhaust commencing at or about bottom dead centre and terminating not later than the closure of the exhaust port of the engine.

5. In a two stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port, an exhaust duct leading from the exhaust port, of means for introducing a quantity of fresh gases into the exhaust system while the exhaust port is still open and at a point in the exhaust duct situated nearer the cylinder than the point in this duct from which the burnt gases return after their mass exit from the cylinder, the said introduction being effected after the mass exit of the burnt gases from the cylinder and before the return of these gases to the cylinder.

6. In a two stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust duct leading from the exhaust port, of an internal source of compressed gaseous fluid, distribution means connecting said source of fluid to an orifice situated on the duct at a point nearer the cylinder than the limit of outward travel of the burnt gases upon their mass exit from the cylinder, and means for controlling said distribution means for introducing compressed gaseous charge into the exhaust duct through said orifice after the mass exit of the burnt gases from the cylinder and before the return of these gases to the cylinder.

7. In a two stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port and an exhaust duct leading from said exhaust port, and an orifice on said duct situated nearer the cylinder than the limit of outward travel of the burnt gases upon their mass exit from the cylinder, a tubular element within said exhaust duct communicating with said orifice and extending in the direction of exhaust, a source of compressed gaseous fluid, distribution means connecting said source of fluid to said orifice and tubular element, and means for controlling said distribution means for introducing compressed gaseous fluid into the exhaust duct through said tubular element after the mass exit of the burnt gases from the cylinder and before the return of these gases to the cylinder.

8. In a two stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port and an exhaust duct leading from said exhaust port and an orifice on said duct situated nearer the cylinder than the limit of outward travel of the burnt gases upon their mass exit from the cylinder, a nozzle within said exhaust duct communicating with said orifice and extending in the direction of exhaust, a source of compressed gaseous fluid, distribution means connecting said source of fluid to said orifice and nozzle, and means for controlling said distribution means for introducing compressed gaseous fluid into the exhaust duct through said nozzle after the mass exit of the burnt gases from the cylinder and before the return of these gases to the cylinder.

9. In a two stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port, and an exhaust duct leading from the exhaust port of means for introducing the whole of the fresh gases required for the working charge, into the said duct at a point situated nearer the cylinder than the limit of outward travel of the burnt gases upon their mass exit from the cylinder, after the said mass exit has occurred and before the return of the burnt gases to the cylinder.

10. In a two stroke cycle internal combustion engine of the kind described, the combination with a cylinder having an exhaust port and an exhaust duct leading from the exhaust port, of an orifice on said duct close to the cylinder, a source of fluid, distributing means connecting said source of fluid to said orifice and means for controlling said distribution means to introduce a working charge of gaseous fluid into the exhaust duct after the mass exit of the burnt gases from the cylinder and before their return to the cylinder.

MICHEL KADENACY.

Aug. 8, 1939. M. KADENACY 2,168,528

EXHAUST PASSAGE OF TWO-STROKE INTERNAL COMBUSTION ENGINES

Filed Oct. 25, 1935

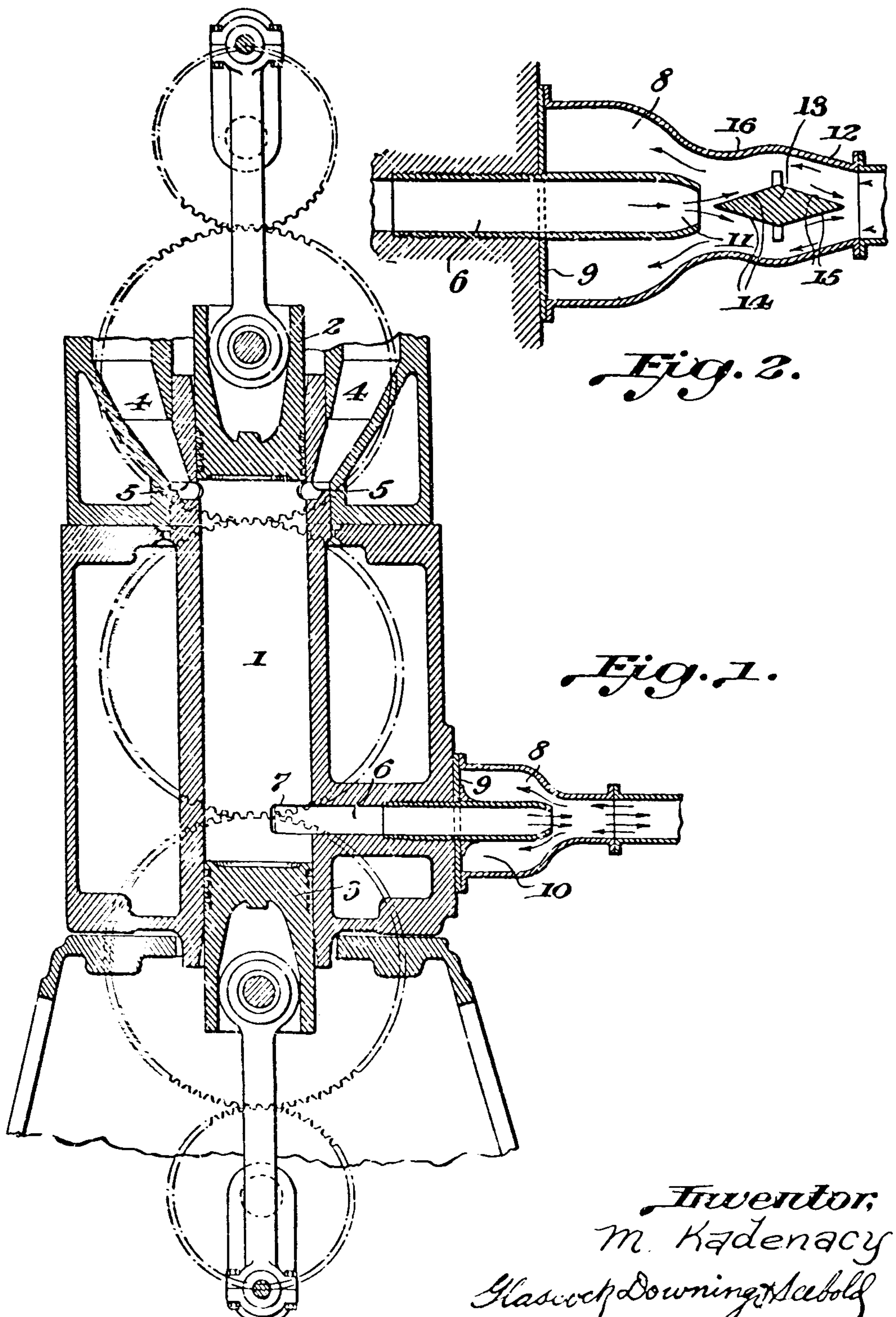

Fig. 2.

Fig. 1.

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Patented Aug. 8, 1939 2,168,528

UNITED STATES PATENT OFFICE

2,168,528

EXHAUST PASSAGE OF TWO-STROKE INTERNAL COMBUSTION ENGINES

Michel Kadenacy, Paris, France

Application October 25, 1935, Serial No. 46,805
In Great Britain November 8, 1934

3 Claims. (Cl. 60—32)

This invention relates to two-stroke cycle internal combustion engines wherein at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when a flow resulting from an adiabatic expansion only is involved, and in such a short interval of time that it is discharged as a mass leaving a depression behind it which is utilised in introducing a fresh charge into the cylinder by opening the inlet orifice with the required delay after the opening of the exhaust orifice to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct and that a suction effect is exerted at the inlet orifice as a consequence of the exit of the said mass.

The applicant has explained, for example in Patents numbered 2,102,559 issued December 14, 1937; 2,110,986 issued March 15, 1938 and 2,113,480 issued April 5, 1938, that he has found that in an internal combustion engine, the gases in the cylinder when exhaust opens behave as though they formed a resilient body, possessing ballistic energy.

When the exhaust orifice first commences to open there is a period of delay during which no appreciable movement of the static gases external to the exhaust orifice can be observed, and after this delay has elapsed the burnt gases issue as a mass at high velocity, and by passing through the exhaust orifice form a column moving rapidly in the direction of exhaust.

Thereafter this outward movement is reversed in direction, and a return of the gases towards the cylinder takes place.

When the burnt gases leave the cylinder as a mass in the manner stated above, they leave a depression behind them in the cylinder and in the exhaust duct close to the cylinder, the volume of which will depend upon the distance to which the mass of burnt gases travels away from the cylinder. Consequently, if the inlet is opened while the exhaust orifice is open and the burnt gases are moving outwardly through the exhaust orifice or duct, the charge will enter into a substantially void space and may, therefore, if desired be introduced by atmospheric pressure alone.

In this connection it should be noted that the mass exit may be considered to have commenced when the exhaust gases have acquired such a momentum in the direction of exhaust that if the inlet orifice is then opened they will continue to move in the direction of exhaust and will not leave the cylinder through the open inlet. This may, for example, be determined by means of records of pressures obtained at the inlet and exhaust ducts.

As, owing to mechanical limitations, the inlet orifice can be opened only progressively, and not instantaneously, it will be understood that it is of advantage to commence the opening of the inlet orifice at an earlier instant of time than that at which the suction effect is exerted thereat, so that when the suction effect is exerted, a sufficient area of inlet orifice is opened to permit optimum utilisation thereof.

But the return of the burnt gases, if it occurs too soon and while inlet and exhaust are still open, materially affects the charging. If at a certain speed, in a variable speed engine, the return coincides with the closure of inlet, which is a satisfactory condition, then at all lower speeds, the return will occur at an earlier crank angle.

This will tend to foul the charge and will also reduce the period of depression and the effect on the torque will be to cause it to fall rapidly away from its optimum value.

Further in a fixed speed engine, if the exhaust pipe is changed for one having characteristics such that an earlier return of the burnt gases occurs, this will have the same effect.

The object of the invention is to provide means in the exhaust passages of such engines whereby the effect of such an early return of the burnt gases before the closure of inlet is minimised, always assuming exhaust to be open when this return occurs.

With this object in view the invention consists in providing means in the exhaust duct preferably close to the engine cylinder whereby a free outflow of the exhaust gases in the direction of exhaust is permitted and any return of the said gases tends to be by-passed into a chamber other than the engine cylinder.

In this way the torque of the engine is rendered more stable under the varying conditions described above.

In establishing the position of such means relative to the engine cylinder it should be borne in mind that the burnt gases return from some zone along the exhaust pipe which may be determined by taking records at intervals along the exhaust pipe, by apparatus capable of revealing the direction of motion, velocity and pressure of the gases during the exhaust period. It will be obvious that if the means according to the invention are situated further from the engine cylinder than the point from which this return occurs, they will be inoperative.

Preferably such means should be situated close

to the cylinder, but a certain amount of latitude may be allowed within the limits defined above.

In a practical embodiment of such an arrangement the exhaust pipe terminates at its end connected to the cylinder in an enlarged chamber preferably of bulbous form and the exhaust gases are fed past this chamber through a relatively short neck or nozzle, the outlet cross-section of which is small relative to the cross-section of the chamber at this point.

In a modification of such an embodiment a deflector body is mounted in the exhaust duct in such a way that it allows the passage of the explosition gases issuing from the neck or nozzle but opposes the return of gases into the cylinder through this nozzle, as described in Patent No. 2,110,986 and deflects any return of the explosion. gases into the said chamber.

As stated above, the device according to the invention is provided in order to counteract any objectionable effect of an early return of the burnt gases on the charging.

Further, an increase in speed of the engine, or a change in exhaust pipe may cause the return of the burnt gases to occur after exhaust closes, so that exhaust closes when the depression still exists in the exhaust duct. If in such an engine inlet is arranged to close before exhaust closes, this will cause a loss of charge from the cylinder by suction through the exhaust port during the period elapsing between inlet closure and exhaust closure. This again will be represented by a rapid drop in torque beyond the optimum point.

This objection may be avoided by closing exhaust at the same time as or before the closure of inlet.

An engine having such timing and provided with means according to the invention will be protected against the objectionable actions of an early return of the burnt gases at low speeds and a prolonged suction in the exhaust duct at high speeds, and it will also be protected against the disturbing effect of changes in the exhaust pipe.

The invention will now be described simply by way of example, and with reference to the accompanying drawing, in which:

Figure 1 is a cross sectional view through the cylinder of an opposed piston engine, the exhaust passage of which is provided with an example of a device according to the invention.

Figure 2 illustrates a modification of the device applied to the exhaust pipe in Figure 1.

In Figure 1 the invention is illustrated as applied to an opposed piston engine comprising a cylinder 1, in which move two opposed pistons 2 and 3. The piston 2 controls the inlet ports 5 communicating through ducts 4 with a source of fresh air or gas, for example the external atmosphere, and the piston 3 controls the exhaust port 7 which discharges the burnt gases from the cylinder through the duct 6, the end of which situated adjacent the cylinder is provided with means in accordance with the invention.

With an engine of the kind illustrated it is well known that while retaining the piston control of the ports, inlet may be arranged to close before or after exhaust closes by suitably off-setting the cranks of the two pistons when assembling the gears.

According to the invention, the exhaust pipe 6 is enlarged at its end connected to or closely adjacent the cylinder to form a chamber 8 of bulbous shape, having on its side facing the exhaust pipe an annular wall 9 which in the example is shown as being curved inwards at its centre orifice so as to give the chamber a toroidal shape, but which may be flat as shown in Figure 2.

A short neck or nozzle 10 is mounted or formed upon the exhaust port and extends coaxially into this charmber 8, while carrying the wall 9 of the latter, and the diameters of the chamber and nozzle on a plane including the outlet end of the nozzle are so proportioned that the cross sectional area of the chamber 8 at this point is great compared with that of the nozzle.

With the arrangement shown in Figure 1, when the burnt gases leave the cylinder and issue from the nozzle they leave behind them a depression which causes the chamber 8 to be evacuated. Upon the opening of inlet, the charge admitted into the cylinder reduces the depression existing therein, and this charge tends to follow the path of the burnt gases through the exhaust port and through the nozzle. On account of the suitable arrangement of the outlet end of the nozzle in the throat of the chamber, any portion of the charge which issues through the nozzle will have the minimum tendency to enter the chamber, so that the depression in the latter will tend to be maintained at a higher level than that in the cylinder. Consequently when the burnt gases return they will not only find an easier path of entry into the chamber on account of the larger area of the annular space surrounding the nozzle, but they will also tend to enter the chamber 8 in preference to the cylinder on account of the fact that a higher depression exists at this moment in the chamber.

The requirement to be borne in mind in selecting suitable proportions for the outlet area of the nozzle and the area of the annular space surrounding this outlet is that when the burnt gases return they should find a less restricted path through the annular space than through the nozzle.

In practice it has been found that a suitable relationship is obtained if the area of the annular space is twice that of the nozzle outlet.

In co-operation with the relative areas, the nozzle outlet should be so positioned that the issuing gases upon leaving the nozzle exert the maximum influence in evacuating the chamber 8. It will therefore be seen that the annular space should not be excessively great in proportion to the nozzle outlet area, and that the nozzle outlet should be suitably situated with respect to the outlet end of the chamber.

In establishing the position of the nozzle outlet relative to the chamber, it should also be noted that if the nozzle is extremely short then the chamber will become a simple expansion chamber while if the nozzle extends too far into the exhaust pipe it may prevent the entry of the returning burnt gases into the chamber.

A suitable arrangement is obtained when the outlet end of the nozzle is situated at or about the throat of the chamber 8, as indicated in the figure. In practice it is found that a little movement from this position does not reduce to any considerable extent the output of the engine, but that as the nozzle outlet is moved further from the throat of the chamber and nearer the cylinder, a point of instability is reached after which the torque drops suddenly to a lower value; thereafter further movement of the nozzle outlet in the same direction produces a further and progressive decrease in output.

If records are taken on the exhaust pipe of an engine in the manner indicated above, both with

and without the device according to the invention, it will be found that when the device is fitted the return of the gases is delayed and also that the peak of pressure accompanying such return is considerably reduced. It will also be found that the moment of outflow of the burnt gases remains substantially unaffected, so that the timing of inlet remains satisfactory and the duration of the suction into the cylinder is extended.

The device according to the invention may advantageously be combined with means which oppose the re-entry of the burnt gases through the neck or nozzle into the cylinder while permitting them to enter the chamber, as illustrated in Figure 2.

The arrangement illustrated in Figure 2 is similar to that illustrated in Figure 1 as regards the general form and arrangement of a nozzle 11 delivering into a chamber 12, but in this embodiment the device described above is combined with deflecting means situated in the exhaust duct and adapted to permit the outflow of the burnt gases and to prevent their re-entry through the neck or nozzle into the cylinder by deflecting the returning gases into the chamber 12.

A deflector body 13 having the shape of two cones with their bases in contact is mounted in the exhaust pipe in alignment with the nozzle 11 and preferably close to the outlet end of the latter. This deflector body is advantageously so arranged that its surface 14 allows the passage of the explosion gases in the outflowing direction, while its surface 15 guides any return wave of the gases into the chamber, the wall of the chamber 12 around the deflector being suitably shaped for this purpose as indicated at 16.

This deflector body is only illustrated by way of example and it will be understood that any other suitable deflecting means may be employed.

By the addition of the deflector body or the like a highly efficient obstacle is presented to any return into the cylinder of explosion gases.

It should be noted that the action which creates the suction effect occurring in engines according to the present invention originates in the cylinder and is propagated from the cylinder into the exhaust duct, in that this suction effect is caused by the exit of at least a substantial portion of the burnt gases from the cylinder at a speed greatly in excess of the speed obtaining when a flow resulting from an adiabatic expansion only is involved and in such a short interval of time that it is discharged as a mass. In carrying out the present invention the natural tendency of the burnt gases to project themselves from the cylinder as a mass should be facilitated and not opposed, that is to say the area of the exhaust orifice available for the discharge of the burnt gases should be as large as possible and the interval of time in which the area required for this discharge of the burnt gases is made available should be as short as possible in order to obtain the most satisfactory results.

I claim:

1. A two-stroke cycle internal combustion engine, wherein the burnt gases are discharged from the cylinder into an exhaust conduit substantially as a mass whereby the said mass moves outward and thereafter returns towards the cylinder from a point which may be within the said exhaust conduit, the said conduit providing a free passage for the burnt gases to the limit of outward travel of said mass, and wherein an inlet is opened for the introduction of fresh charge while the exhaust port is still open and when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, having means on the exhaust system for modifying the action of the said mass on the gases in the exhaust pipe, and the action of the return of the burnt gases upon the cylinder contents, the said means comprising a chamber other than the engine cylinder in open communication with the exhaust conduit at a point situated nearer the cylinder than the limit of outward travel of the burnt gases upon their mass exit from the cylinder, whereby upon the opening of the exhaust orifice for the discharge of the mass of burnt gases, the said mass is directed past the said communication during its outward motion, so that there is first an increase in pressure in the exhaust pipe adjacent the cylinder, which is transmitted through the gases contained in the exhaust pipe to the gases contained in said chamber, and thereafter a depression is formed in the cylinder and in the exhaust pipe adjacent the cylinder followed by a depression in the said chamber, and returning gases enter the said chamber, which is then at a lower pressure than the engine cylinder and the exhaust pipe adjacent the engine cylinder.

2. A two-stroke cycle internal combustion engine wherein the burnt gases are discharged from the cylinder into an exhaust conduit substantially as a mass whereby the said mass moves outward and thereafter returns towards the cylinder from a point which may be within the said exhaust conduit, the said conduit providing a free passage for the burnt gases to the limit of outward travel of said mass, and wherein an inlet is opened for the introduction of fresh charge while the exhaust port is still open and when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, having a chamber other than the engine cylinder in open communication with the exhaust conduit at a point situated nearer the cylinder than the limit of outward travel of the burnt gases upon their mass exit from the cylinder and means forming a part of the said conduit whereby the said mass is directed past the said communication during its outward motion and returning gases enter said chamber.

3. A two-stroke cycle internal combustion engine as claimed in claim 1, wherein the said chamber is constituted by a bulbous enlargement of the exhaust pipe at its end closely adjacent the cylinder and a short neck or nozzle is formed or mounted upon the exhaust port so as to extend coaxially substantially into the throat of this chamber.

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Nov. 1, 1938. M. KADENACY 2,134,920

INTERNAL COMBUSTION ENGINE

Filed Jan. 23, 1936 2 Sheets-Sheet 1

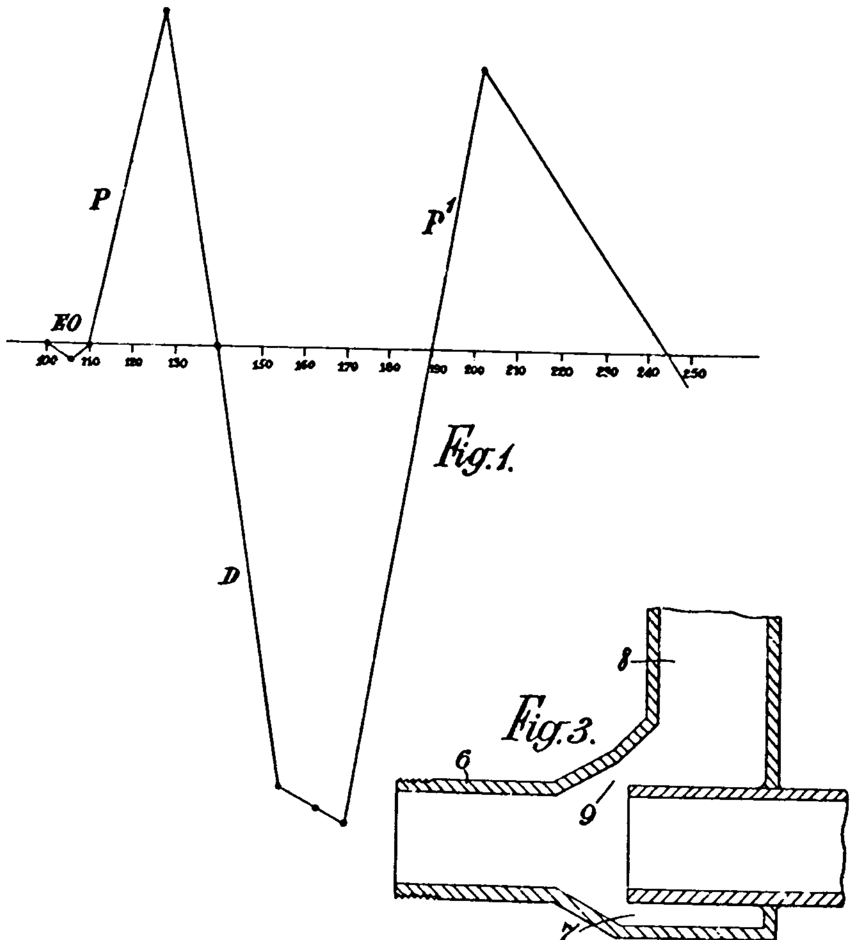

Fig. 1.

Fig. 3.

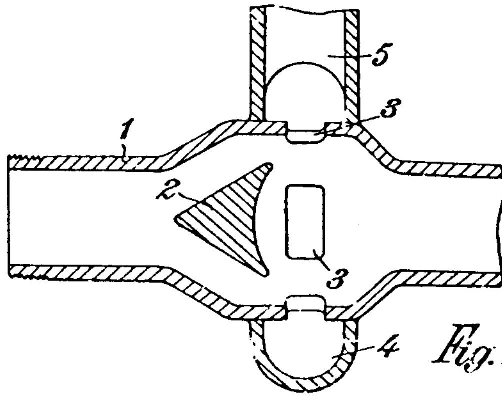

Fig. 2.

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INTERNAL COMBUSTION ENGINE

Filed Jan. 23, 1936 2 Sheets-Sheet 2

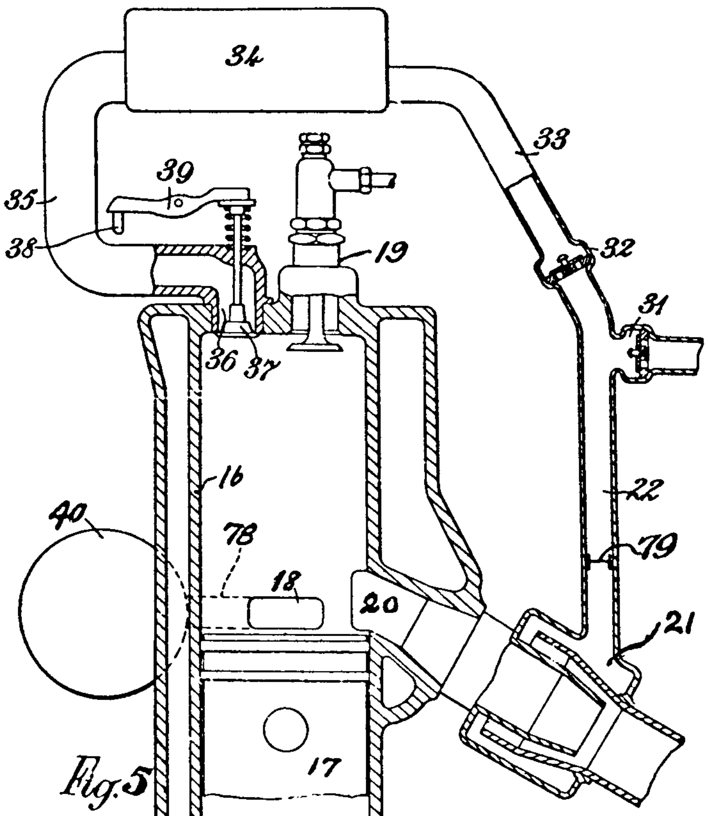
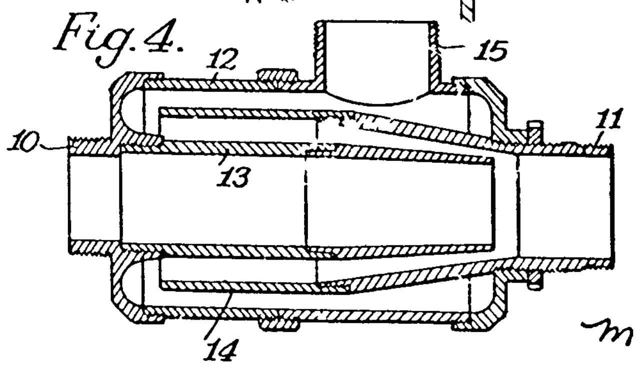


Fig. 5

Fig. 4.



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Patented Nov. 1, 1938 2,134,920

UNITED STATES PATENT OFFICE

2,134,920

INTERNAL COMBUSTION ENGINE

Michel Kadenacy, Paris, France

Application January 23, 1936, Serial No. 60,529
In Great Britain February 11, 1935

38 Claims. (Cl. 123—65)

The applicant has found that in an internal combustion engine, the behaviour of the gases is such as to lead to the conclusion that as a consequence of the combustion of the charge, the burnt gases form a mass having a high initial velocity and possessing properties similar to those of a resilient body, so that when the exhaust orifice opens this mass seeks to project itself bodily from the cylinder and to leave the latter in a consequent vacuous condition.

The applicant has already proposed to utilize the void left in the cylinder when the burnt gases are discharged from the cylinder into the exhaust system as a mass, for the purpose of introducing a fresh charge into the cylinder.

In the operation of such engines the applicant has found that the burnt gases do not leave the cylinder immediately the exhaust orifice commences to open. There is first a period of delay, during which the burnt gases do not issue from the cylinder and after this delay has elapsed the burnt gases issue bodily from the cylinder with an extremely high velocity as a mass which responds to the laws of reflection and rebound and it leaves in the cylinder a profound depression. Subsequently, this outward motion of the burnt gases is reversed in direction and if the gases are allowed to re-enter the cylinder they destroy the depression left therein.

Accordingly the applicant has already proposed a method of charging two-stroke cycle internal combustion engines which consists in opening the admission orifice for the introduction of a fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust system as a consequence of their mass exit from the cylinder.

The present invention relates to such engines, but more generally to internal combustion engines or machines wherein the burnt gases, issue from the explosion chamber as a mass at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber.

In such engines or machines as a consequence of each exhaust operation, when the burnt gases are discharged through an exhaust duct, two pressure or impulse phases will be produced in the exhaust duct as a consequence of the mass exit and return of the burnt gases, and one depression or suction phase will be produced in the exhaust duct after the mass of burnt gases has left the cylinder and before it returns to the latter.

The intensity of the depression left in the working chamber by the exhaust gases when they leave the latter in mass form is very great. This depression exists in the cylinder and also the exhaust duct in a space which may be several times greater than the volume of the cylinder.

The invention consists in a method of utilizing an internal combustion engine or machine wherein the burnt gases are evacuated from the working chamber as a mass at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber for producing a flow of fluid external to the said engine, consisting in employing the periods of depression and impulse or pressure left and produced in the exhaust duct of the said engine by the mass exit of the burnt gases from the cylinder into the exhaust system in order to aspirate the said fluid and deliver the aspirated fluid to a point of utilization, disposal or storage.

The invention further consists in the combination of an internal combustion engine or machine, wherein the burnt gases are evacuated from the working chamber of the engine or machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber with a pump having suction and delivery orifices and a communication with the interior of the exhaust duct, at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, the arrangement being such that the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the mass exit of the burnt gases and/or the return of the said mass towards the cylinder initiates a delivery stroke or strokes of the said pump.

In a practical embodiment of the invention the engine to which the invention is applied is a two-stroke cycle internal combustion engine of the kind wherein the void left in the cylinder by the mass exit of the burnt gases from the cylinder into the exhaust duct is utilized in charging the cylinder, by opening an inlet for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the cylinder.

The action which creates the suction effect occurring in engines according to the present invention originates in the working chamber in which the combustion has been effected and is propagated from this chamber into the exhaust duct and into the pump, in that this suction effect is caused by the exit of the burnt gases from the working chamber at a speed greatly in excess of that obtaining when adiabatic action only is involved and in such a short interval of time that it is discharged as a mass.

In carrying out the present invention the natural tendency of the burnt gases to project themselves from the cylinder as a mass should be facilitated and not opposed, that is to say the area of the exhaust orifice available for the discharge of the burnt gases should be as large as possible and the interval of time in which the area required for this discharge of the burnt gases is made available should be as short as possible, in order to obtain the most satisfactory results.

As stated above the effects utilized in carrying out the present invention originate in the working chamber and are not produced by any action exerted by the exhaust duct but in carrying out the invention the exhaust duct should be of such a form that it permits the utilization of the actions in question. The exhaust pipe between the working chamber and the pump provided upon the exhaust duct should be free from abrupt restrictions or enlargement of cross-section and the connection between the pump and the exhaust duct should be situated at a point nearer the working chamber than the point of return of the burnt gases in the exhaust duct.

Some embodiments of the invention will now be described, simply by way of example, and with reference to the accompanying drawings, in which:

Figure 1 is a curve of pressures and depressions taken in the exhaust duct of an engine during the exhaust period.

Figures 2, 3 and 4 illustrate examples of exhaust duct arrangements whereby the depression left in the exhaust duct may be utilized in order to aspirate a fluid, and the pressure impulses occurring therein may also be utilized to deliver this aspirated fluid.

Figure 5 shows an embodiment, in which the invention is employed for supplying a supplementary charge of air to a two-stroke engine.

In Figure 3 the delivery is chiefly obtained by the direct impulse produced by the explosion gases upon issuing from the cylinder.

In Figure 4 the delivery is chiefly effected by the return impulse of the discharged exhaust gases, and in Figure 2 both the direct impulse and the return impulse of the exhaust gases are utilized.

If a record is taken of the pressure variations in the exhaust pipe of an internal combustion engine of the kind referred to during the exhaust period, a curve similar to that shown in Figure 1 may be obtained, in which EO represents the opening of exhaust, the ordinates represent pressures above and below atmospheric pressure and the abscissae crank angles in degrees.

It should be mentioned that in the figure the pressures above and below atmospheric pressure are not shown in scale relationship.

Such a curve may be obtained for example by utilizing a stroboscopic device formed by a ported tube mounted in or on the exhaust duct and rotating with the engine and a stationary but angularly adjustable ported sleeve on this tube having its port connected to a manometer, a pressure or depression impulse being obtained each time the port in the tube and sleeve coincide and a record being taken when a steady reading is given on the manometer, the crank angle at which each reading is taken being determined by the angular adjustment of the sleeve.

The curve shows clearly the two phases of pressure or impulse P, P' that occur during the outflow and return of the gases and the intervening phase of depression D. This curve is characteristic for all internal combustion engines of the kind referred to but the moments at which the outflow and return of the gases occur will vary.

The moments during the operation of the engine at which the aspiration of fluid and the delivery of such aspirated fluid may be produced in accordance with the invention, will be seen from Figure 1.

In the first place, a delivery phase may be produced by the direct shock or impulse of the issuing exhaust gases which occurs very shortly after the opening of exhaust, as shown by the part P of the curve.

A depression is then formed in the cylinder and a little later a depression D is formed in the exhaust duct and the intensity and magnitude of the volume in which it exists will be proportionate to the kinetic energy contained in the exhaust gases.

At this moment the aspiration of external fluid through the medium of a communication with the interior of the exhaust duct may be obtained

The return shock or impulse P' follows and destroys this depression phase and may constitute a second delivery phase.

The cycle may thus be considered to occur in the following manner: aspiration of fluid on account of the depression existing in the exhaust duct; then the delivery of the aspirated fluid by the return impulse of the discharged gases which follows this depression and after a rotation of the crank through about 280°, a second delivery by the direct impulse of the burnt gases from the following explosion.

It should be noted that the intensity of the phenomena described above is inversely proportional to the distance from the cylinder. Consequently, the communication with the interior of the exhaust duct should be located close to the cylinder.

In prior British specifications, the applicant has described internal combustion engines in which it is proposed to utilize the vacuum or high depression left in the cylinder by the explosion gases when they issue from the cylinder as a mass in order to introduce a fresh charge through the main admission ports.

In an engine of this kind, the fresh charge admitted to the cylinder cannot fill the complete cavity left in the cylinder and in the exhaust duct by the issuing gases for practical reasons depending upon the position, shape and surface of the admission orifices.

In fact, the applicant has found that if an orifice is provided in the exhaust duct close to the cylinder and is opened to a source of gaseous fluid external to the exhaust duct, the volume of gaseous fluid drawn directly into the exhaust duct through this orifice during the depression phase described above does not in any way impede or reduce the charge admitted directly into the cylinder through the usual admission orifices,

although the volume of gaseous fluid drawn into the exhaust duct may be equal to this charge.

Consequently by providing a chamber communicating with the interior of the exhaust duct of an internal combustion engine, and with a source of fluid external to the exhaust duct, this fluid can be aspirated into the said chamber and if desired it can be delivered into a receiver, where it may be stored under a suitable or chosen pressure and from which it may be utilized for supplying the engine or for any required purpose, and if desired the depression left in the cylinder by the issuing exhaust gases may still be utilized in order to introduce a fresh charge through the main admission ports.

Suitable distribution means must be provided so that the aspirated air or other gaseous fluid will follow the required direction during the suction and delivery periods, and so that it cannot return from the paths it is required to follow.

These means may for example consist of non-return valves or of controlled valves or of means such as those described in British Specifications Nos. 35069/33 and 25165/34.

The communication with the interior of the exhaust duct must be suitably arranged to permit a utilization of the depression and pressure phases in accordance with the invention, the communication may be so arranged that use is made chiefly of the return impulse of the exhaust gases or of both these impulses one after the other, for the purpose of delivering and/or compressing the aspirated charge.

Figures 2, 3 and 4 illustrate three examples of arrangements of the communication with the interior of the exhaust duct.

Figure 2 illustrates an arrangement of intake which makes use of means such as those described in the applicant's prior British Specification No. 35069/33. In this figure, which may be considered as a section through the exhaust duct of an internal combustion engine, the duct 1 is enlarged in order to receive a cone shaped obturator 2, the point of which will face the cylinder and the concave base of which will be turned towards the outlet end of the exhaust pipe, this obturator being arranged in relation to the walls of the duct in such a manner as to permit the outflow of the burnt gases and to prevent a return wave of these gases from re-entering the cylinder.

The part of the duct receiving the obturator 2 is provided with orifices 3, communicating with an annular chamber 4 provided or formed around the duct 1, and having an outlet 5 for connection to a source of fluid external to the duct 1.

In this example, the intake orifices 3 for the aspirated fluid are arranged so that the direct impulse and the return impulse of the exhaust gases will be transmitted in such a way that the aspirated gases may be delivered by the highest intensity of each of these two compression agents although it will be understood that the intensity of the direct impulse will always be higher than that of the return impulse.

Figure 3 illustrates a form of intake which ensures that the direct impulse will have the greatest action in delivering the aspirated charge.

In this figure, the exhaust 6 is surrounded by an annular chamber 7 having an outlet 8 and the duct is interrupted by an annular space 9 establishing a communication between the chamber 7 and the interior of the duct 6. The walls of the chamber 7 are flared in continuation of the walls of the duct on the side for connection to the cylinder so as to ensure that the direct impulse of the exhaust gases will be transmitted into this chamber.

A practical form of intake which chiefly facilitates the utilization of the return impulse, and which has given satisfactory results in practice is illustrated in Figure 4.

In this figure the exhaust duct is formed by two portions 10 and 11 connected together by a chamber 12. The portion 10 of the duct is extended into the said chamber by means of a tubular element 13 opening into the portion 11 and situated in the interior of a tubular element 14 which extends the portion 11 of the duct and stops short of the inner wall of the said chamber 12. The annular space left between the elements 13 and 14 establishes a communication between the interior of the exhaust duct and the chamber 12.

The portion 11 of the duct is adjustably connected with the chamber 12 in order to permit a regulation of the distance between the free end of the element 14 and the internal wall of the chamber situated towards the cylinder.

The diameter of the free end of the element 13 is slightly smaller than that of the duct 11 and the element 14 at this point is slightly flared so as to provide a passage of increasing section between the elements 13 and 14 from the free end of the element 13.

The length of the element 13 may also be regulated by means of the screw connection provided between this element and the duct 10. These adjustments enable the action of the device to be varied so as to vary the relation between the suction and the delivery of fluid and the intensity of these actions which will be exerted through the outlet 15 of the chamber 12.

Figure 5 shows an engine arrangement in which the aspirated fluid is drawn from the atmosphere and the pressure impulses in the exhaust duct are utilized in order to deliver this aspirated air to a supplementary inlet port on the cylinder.

This figure shows an engine cylinder 16 in which moves a piston 17. Air is admitted by atmospheric pressure through the inlet 18 and fuel is introduced by the injector 19. Exhaust takes place through the duct 20. Upon the exhaust duct close to the cylinder is provided an intake 21, which, by way of example, is shown similar to that illustrated in Figure 4 communicating with a first chamber 22.

It is of advantage for the chamber 22 to be of tubular form because the aspirated charge and the exhaust gases that will deliver this charge will be in contact with each other and it will be of interest to prevent them from mixing.

The chamber 22 is provided at its end remote from the exhaust duct with a suction valve 31 or the like communicating with an external source of fluid, which in the example will be the air of the atmosphere, and a delivery valve 32 or the like. This delivery valve is followed by chambers, which may according to requirements be tubular or in the form of reservoirs or may be simply ducts for storing, leading or presenting the aspirated and delivered gas to its point of utilization.

In the example, the valve 32 leads through a duct 33 to a reservoir 34 itself connected to a duct 35 which leads the aspirated and delivered gas to a point of utilization.

The operation of this apparatus may be compared to a piston pump in which the function of the piston is taken by the exhaust gases while in mass form in the exhaust duct which exhaust

the chamber 22 during what may be considered as a downward stroke of the piston and then deliver the charge thus aspirated by means of what may be considered as two upward strokes of the piston.

An apparatus such as that described above may be employed for example as a compressor and the compressed charge thus obtained may be utilized for supplying a charge of air or combustible gas to the engine cylinder.

In the example, the duct 35 is put into communication with a supplementary admission orifice 36 provided upon the cylinder and serves for supplying an additional charge of air under pressure to the engine, the main charge being admitted by atmospheric pressure through the orifice 18. This additional charge may, for example be introduced at the end of the main admission in order to serve as correcting air, as described in the applicant's prior British Specification No. 24372/34, or it may serve for supplying a charge at any chosen moment through a suitable distribution means. In this case a great advantage is obtained as compared with the use of a compressor which draws power from the engine, is more costly, and complicates the construction.

The orifice 36 is controlled by any suitable means, for example by a valve 37, operated by push rod 38 and rocker arm 39 as indicated diagrammatically in the figure, in order to open at the chosen moment during the cycle of operations of the engine.

If the delivery valve 32 is omitted and the air is delivered directly to the admission valve 37, the engine will work at a more or less fixed speed because in this case the moment of opening of the valve 37 will have to coincide with a moment at which a charge is delivered from the chamber 22. But in the case when the chamber 22 delivers into the reservoir 34 through a non-return valve or equivalent means, the engine will draw its charge from the reservoir 34 at the required moment and its speed will become independent of the moments at which the suction and delivery stages occur in the chamber 22.

By suitably proportioning the chambers 22, 33, 34 and 35, the charge delivered from the chamber 22 may be stored or supplied to the point of utilization at or above atmospheric pressure.

In the example described with reference to Figure 5, instead of supplying the main charge to the engine by atmospheric pressure, this main charge may be introduced by any suitable means, for example by a compressor 40, connected to the inlet port 18 by a duct 78 as indicated in Figure 5.

It should be noted that as shown in Figure 5, a piston element such as a light, freely movable disc 79 may be arranged in the chamber 22, without in any way affecting the principle of operation of the device.

The invention is applicable to engines or machines having any number of strokes per cycle, but as only two strokes are essential to complete the cycle, the invention will most advantageously be applied to two stroke cycle engines in which the charge is introduced, at or above atmospheric pressure, into the combustion chamber by utilizing the phenomena described.

The valves employed for the suction and for the delivery of an aspirated fluid in an arrangement according to the invention may be simple or multiple. These valves may be arranged so as to provide a passage of large area and they may have any shape provided they respond to rapid suctions and equally rapid deliveries.

As stated above, these valves may be replaced by deflectors such as those described in the applicant's prior British Specification No. 35069/33, but in this case the delivered fluid cannot be stored under compression.

In this last example, it may be imagined that the delivery pulsations of fresh air or of the fresh charge are synchronized with the appropriate moments for the introduction of these gases into the engine cylinder, so that the inlet opening for these gases will coincide with the moment at which a high pressure exists in the ducts that deliver the gases, which pressure is produced by one of the delivery agents described above, that is to say, either by the direct impulse or the return impulse of the exhaust gases after their reflection from the atmosphere external to the cylinder.

I claim:

1. A method for producing a flow of fluid external to an internal combustion engine, consisting in employing the periods of depression and impulse or pressure left and produced in the exhaust duct of the said engine by the mass exit of the burnt gases from the cylinder into the exhaust system, in order to aspirate the said fluid, compress this aspirated fluid and if desired deliver the aspirated fluid to a reservoir which may be the engine cylinder.

2. A method for producing a flow of fluid external to an internal combustion machine wherein the burnt gases are evacuated from the working chamber as a mass at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, consisting in employing the periods of depression and impulse or pressure left and produced in the exhaust duct of the said machine by the mass exit of the burnt gases from the working chamber into the exhaust system in order to aspirate the said fluid and deliver the aspirated fluid to a point of utilization, disposal, or storage.

3. A method for producing a flow of fluid external to an internal combustion engine wherein the burnt gases are evacuated from the working chamber as a mass at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, consisting in employing the periods of depression and impulse or pressure left and produced in the exhaust duct of the said engine by the mass exit of the burnt gases from the working chamber into the exhaust system in order to aspirate the said fluid and deliver the aspirated fluid to a point of utilization, disposal, or storage.

4. The combination with an internal combustion machine comprising a working chamber and an exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices, and a communication with the interior of the exhaust duct, at a point situated near the working chamber than the point of return of the burnt gases within the exhaust duct, the arrangement being such that the depression left in the exhaust duct

by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump.

5. The combination with an internal combustion engine, a working chamber, an exhaust orifice and an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and wherein the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, means to open said inlet orifice for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices and a communication with the interior of the exhaust duct, at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, the arrangement being such that the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump.

6. A combination as claimed in claim 4, including a source of gaseous fluid, a conduit connecting said source to the suction orifice of the pump, means for utilizing and storing said fluid, and a conduit connecting said means with the delivery orifice of the pump and wherein the pump includes a chamber having at one end said communication with the exhaust duct and at points remote from the exhaust duct said suction orifice and said delivery orifice.

7. A combination as claimed in claim 4, including a source of gaseous fluid, a conduit connecting said source to the suction orifice of the pump, means for utilizing and storing said fluid, and a conduit connecting said means with the delivery orifice of the pump, and wherein the pump includes a chamber of tubular form having at one end said communication with the exhaust duct and at points remote from the exhaust duct, said suction orifice and said delivery orifice.

8. The combination with an internal combustion machine a working chamber, and an exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices and a communication with the interior of the exhaust duct, at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and means arranged in the exhaust duct in such a way relative to the communication between the exhaust duct and the said pump as to ensure that the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and that both the direct impact and the return impact of the burnt gases will deliver the charge aspirated into the pump.

9. A combination as claimed in claim 8, wherein the said means are formed by deflecting and reflecting surfaces so arranged in the exhaust duct as to permit the free outward passage of the mass of burnt gases and to oppose the return of the said gases to the working chamber while at the same time serving to direct the impacts caused by the said mass of burnt gases on both its outward and return motions into the pump.

10. A combination as claimed in claim 4, including non-return valves for controlling the suction and delivery orifices and opening in the desired direction under the effect of a suction or of a delivery action and closing automatically.

11. A combination as claimed in claim 4, including deflectors for controlling the suction and delivery orifices and permitting the passage of a fluid in one direction and opposing the return of the said fluid.

12. A combination as claimed in claim 4, including distribution means for controlling the suction and delivery orifices so that said orifices open and close at predetermined moments.

13. A combination as claimed in claim 4, wherein the suction orifice communicates with the atmosphere.

14. A combination as claimed in claim 4, including a reservoir and a conduit connecting the delivery orifice of the pump with the said reservoir whereby fluid delivered by the pump may be stored or conveyed or presented to a point of utilization.

15. A combination as claimed in claim 4, including a reservoir and a conduit connecting the delivery orifice of the pump with the said reservoir whereby fluid delivered by the pump may be stored under pressure in said chamber.

16. A combination as claimed in claim 4, including a reservoir, a conduit connecting the delivery orifice of the pump with said reservoir whereby fluid delivered by the pump may be stored under pressure in said reservoir, a supplementary admission orifice upon the working chamber, a conduit connecting said supplementary admission orifice with said reservoir and means controlling said orifice to open said orifice at a chosen moment during the cycle of operations of the machine and put the working chamber into communication with the reservoir.

17. A combination as claimed in claim 5, including a source of gaseous fluid, a conduit connecting said source to the suction orifice of the pump, means for utilizing and storing said fluid and a conduit connecting said means with the delivery orifice of the pump and wherein the pump includes a chamber having at one end said communication with the exhaust duct and at points remote from the exhaust duct said suction orifice and said delivery orifice.

18. A combination as claimed in claim 5, including non-return valves for controlling the suction and delivery orifices and opening in the desired direction under the effect of a suction or of a delivery action and closing automatically.

19. A combination as claimed in claim 5, wherein the suction orifice communicates with the atmosphere.

20. A combination as claimed in claim 5, including a reservoir and a conduit connecting the delivery orifice of the pump with the said reser-

voir whereby fluid delivered by the pump may be stored or conveyed or presented to a point of utilization.

21. A combination as claimed in claim 5, including a reservoir and a conduit connecting the delivery orifice of the pump with the said reservoir whereby fluid delivered by the pump may be stored under pressure in said chamber.

22. A combination as claimed in claim 5, including a reservoir, a conduit connecting the delivery orifice of the pump with said reservoir whereby fluid delivered by the pump may be stored under pressure in said reservoir, a supplementary admission orifice upon the working chamber, a conduit connecting said supplementary admission orifice with said reservoir and means controlling said orifice to open said orifice at a chosen moment during the cycle of operations of the machine and put the working chamber into communication with the reservoir.

23. A combination as claimed in claim 5, including a reservoir, a conduit connecting the delivery orifice of the pump with said reservoir whereby fluid delivered by the pump may be stored under pressure in said reservoir, a supplementary admission orifice upon the working chamber, a conduit connecting said supplementary admission orifice with said reservoir and means controlling said orifice to open said orifice towards the commencement of the main atmospheric admission to put the working chamber into communication with the reservoir and supply a correcting charge of air to the working chamber.

24. A combination as claimed in claim 5, including a reservoir, a conduit connecting the delivery orifice of the pump with said reservoir whereby fluid delivered by the pump may be stored under pressure in said reservoir, a supplementary admission orifice upon the working chamber, a conduit connecting said supplementary admission orifice with said reservoir and means controlling said orifice to open said orifice towards the end of the main atmospheric admission to put the working chamber into communication with the reservoir and supply a correcting charge of air to the working chamber.

25. A combination as claimed in claim 5, including a reservoir, a conduit connecting the delivery orifice of the pump with said reservoir whereby fluid delivered by the pump may be stored under pressure in said reservoir, a supplementary admission orifice upon the working chamber, a conduit connecting said supplementary admission orifice with said reservoir and means controlling said orifice to open said orifice to supply a supercharge to the working chamber after the closure of exhaust.

26. The combination with an internal combustion engine comprising a working chamber, an exhaust orifice, an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and wherein the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, by opening the inlet orifice for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices and a communication with the interior of the exhaust duct, at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and means arranged in the exhaust duct in such a way relative to the communication between the exhaust duct and the said pump as to ensure that the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and that both the direct impact and the return impact of the burnt gases will deliver the charge aspirated into the pump.

27. A combination as claimed in claim 26, wherein the said means are formed by deflecting and reflecting surfaces so arranged in the exhaust duct as to permit the free outward passage of the mass of burnt gases and to oppose the return of the said gases to the working chamber, while at the same time serving to direct the impacts caused by the said mass of burnt gases on both its outward and return motions into the pump.

28. A combination as claimed in claim 4, wherein the exhaust duct comprises two longitudinally displaced portions, the portion nearer the working chamber being of such length that the interruption in the exhaust duct provides an annular space at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including an annular chamber enclosing the exhaust duct at the zone of said interruption, said annular chamber having walls situated towards the working chamber flared in continuation of the walls of the duct and having a communication with said pump, whereby said annular space establishes a communication between the pump and the interior of the exhaust duct via said chamber.

29. The combination with an internal combustion machine comprising a working chamber and exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that the interruption in the exhaust duct provides an annular space at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including an annular chamber enclosing the exhaust duct at the zone of said interruption, said annular chamber having walls situated towards the working chamber flared in continuation of the walls of the duct and having a communication with said pump, whereby said annular space establishes a communication between the pump and the interior of the exhaust duct via said chamber and whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump.

30. The combination with an internal combustion machine comprising a working chamber and

exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump.

31. The combination with an internal combustion machine comprising a working chamber and exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump, means being provided whereby the length of one or the other of the two tubular elements can be varied in order to vary the action of the intake between the pump and the interior of the exhaust duct and/or the relation between the suction and the delivery of external fluid by the pump.

32. The combination with an internal combustion machine comprising a working chamber and exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump, means being provided whereby the length of one or the other of the two tubular elements can be varied in order to vary the action of the intake between the pump and the interior of the exhaust duct and/or the relation between the suction and the delivery of external fluid by the pump, the diameter of the free end of the second tubular element being slightly smaller than that of the first tubular element at this point, and the said first element being flared at this point in order to permit the area of the annular inlet aperture between the two tubular elements to be varied by a relative longitudinal movement of these elements.

33. The combination with an internal combustion machine comprising a working chamber and exhaust orifice, wherein the burnt gases are evacuated from the working chamber of the machine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working cham-

ber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump, means being provided whereby the length of one or the other of the two tubular elements can be varied in order to vary the action of the intake between the pump and the interior of the exhaust duct and/or the relation between the suction and the delivery of external fluid by the pump, the diameter of the free end of the second tubular element being slightly smaller than that of the first tubular element at this point, and the said first element being flared at this point so that the passage between the tubular elements increases in section from the free end of the second element in order to permit the area of the annular inlet aperture between the two tubular elements to be varied by a relative longitudinal movement of these elements.

34. The combination with an internal combustion engine comprising a working chamber, an exhaust orifice and an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and wherein the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, means to open said inlet orifice for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that the interruption in the exhaust duct provides an annular space at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including an annular chamber enclosing the exhaust duct at the zone of said interruption, said annular chamber having walls situated towards the working chamber flared in continuation of the walls of the duct and having a communication with said pump, whereby said annular space establishes a communication between the pump and the interior of the exhaust duct via said chamber and whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump.

35. The combination with an internal combustion comprising a working chamber, an exhaust orifice and an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and wherein the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, means to open said inlet orifice for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by ehe return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump.

36. The combination with an internal combustion engine comprising a working chamber, an exhaust orifice and an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and wherein the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, means to open said inlet orifice for the in-

troduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump, means being provided whereby the length of one or the other of the two tubular elements can be varied in order to vary the action of the intake between the pump and the interior of the exhaust duct and/or the relation between the suction and the delivery of external fluid by the pump.

37. The combination with an internal combustion engine comprising a working chamber, an exhaust orifice and an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and whereby the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, means to open said inlet orifice for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump, means being provided whereby the length of one or the other of the two tubular elements can be varied in order to vary the action of the intake between the pump and the interior of the exhaust duct and/or the relation between the suction and the delivery of external fluid by the pump, the diameter of the free end of the second tubular element being slightly smaller than that of the first tubular element at this point, and the said first element being flared at this point in order to permit the area of the annular inlet aperture between the two tubular elements to be varied by a relative longitudinal movement of these elements.

38. The combination with an internal combustion engine comprising a working chamber, an exhaust orifice and an inlet orifice, wherein the burnt gases are evacuated from the working chamber of the engine through a duct as a mass, at a speed much higher than that obtaining when adiabatic action only is involved and in such a short interval of time that they are discharged wholly or substantially wholly from the working chamber and wherein the void left in the working chamber by the mass exit of the burnt gases from the said chamber into the exhaust duct is utilized in charging the said chamber, means to open said inlet orifice for the introduction of the fresh charge after the exhaust orifice opens, but only with the required delay to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct as a consequence of their mass exit from the working chamber, with a pump having suction and delivery orifices, the exhaust duct comprising two longitudinally displaced portions, the portion nearer the working chamber being of such length that an interruption in the exhaust duct occurs at a point situated nearer the working chamber than the point of return of the burnt gases within the exhaust duct, and including a chamber connecting the two portions of the exhaust duct, an orifice in said chamber, a conduit connecting said orifice to said pump, a first tubular element situated in the interior of said chamber and extending the portion of the exhaust duct more distant from the working chamber towards the working chamber and stopping short of an internal wall of the chamber connecting the exhaust duct portions, and a second tubular element situated within the first tubular element and extending the portion of the exhaust duct nearer the working chamber towards the other portion of the exhaust duct, the space between said tubular elements establishing a communication between the interior of the exhaust duct and the chamber connecting

the exhaust duct portions and thence to the pump whereby the depression left in the exhaust duct by the mass exit of the burnt gases from the working chamber initiates a suction stroke of the said pump and the shocks produced by the return of the said mass towards the cylinder and of the next mass exit of the burnt gases initiate delivery strokes of the said pump, means being provided whereby the length of one or the other of the two tubular elements can be varied in order to vary the action of the intake between the pump and the interior of the exhaust duct and/or the relation between the suction and the delivery of external fluid by the pump, the diameter of the free end of the second tubular element being slightly smaller than that of the first tubular element at this point, and the said first element being flared at this point so that the passage between the tubular elements increases in section from the free end of the second element in order to permit the area of the annular inlet aperture between the two tubular elements to be varied by a relative longitudinal movement of these elements.

MICHEL KADENACY.

Feb. 14, 1939. M. KADENACY 2,147,200

INTERNAL COMBUSTION ENGINE

Original Filed Jan. 23, 1936 2 Sheets-Sheet 1

Fig. 1.

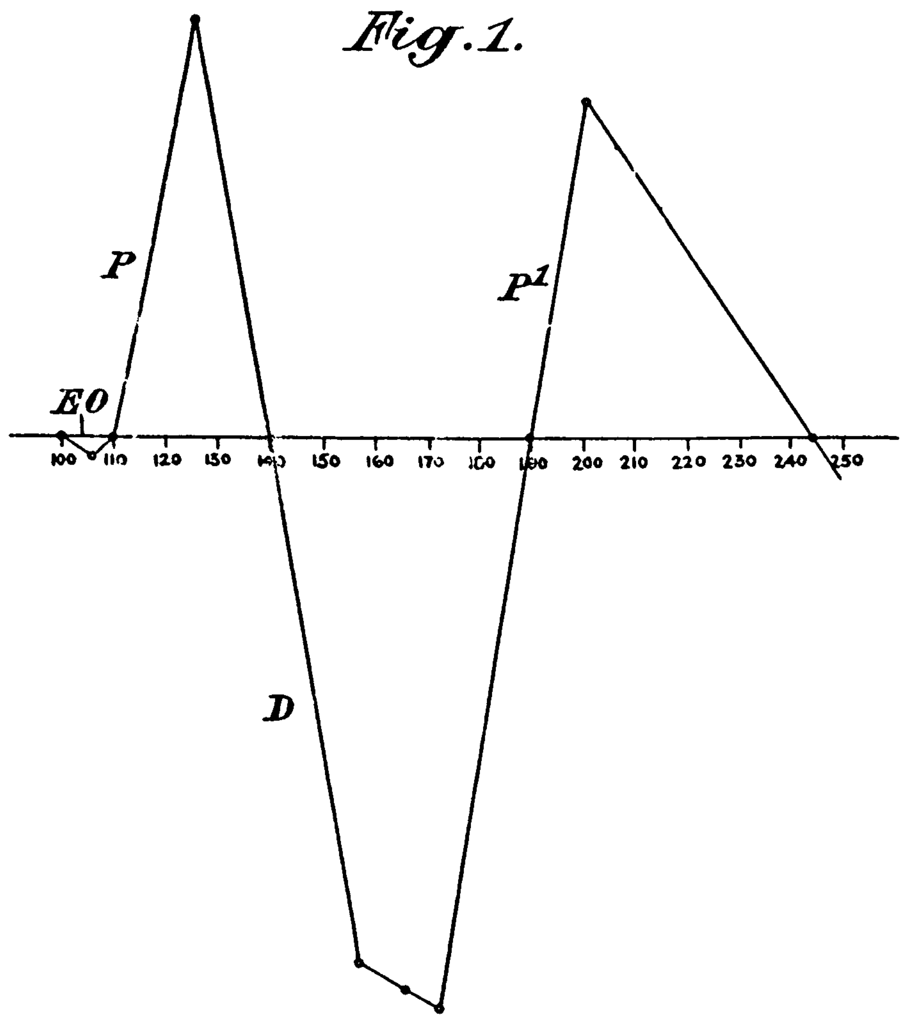
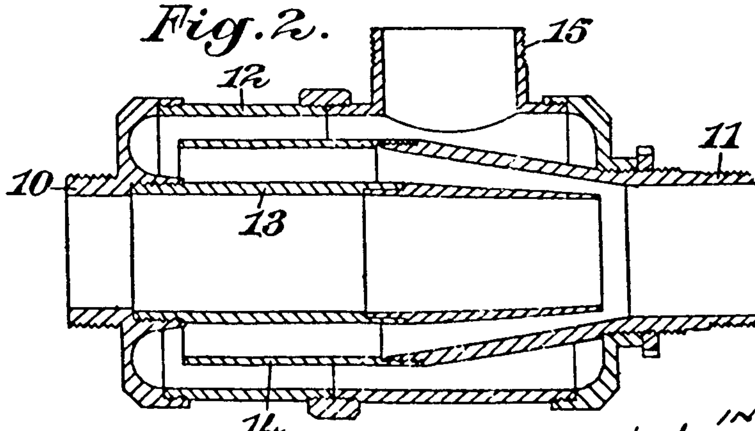


Fig. 2.



INVENTOR:
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Attys.

Feb. 14, 1939. M. KADENACY 2,147,200
INTERNAL COMBUSTION ENGINE
Original Filed Jan. 23, 1936 2 Sheets-Sheet 2

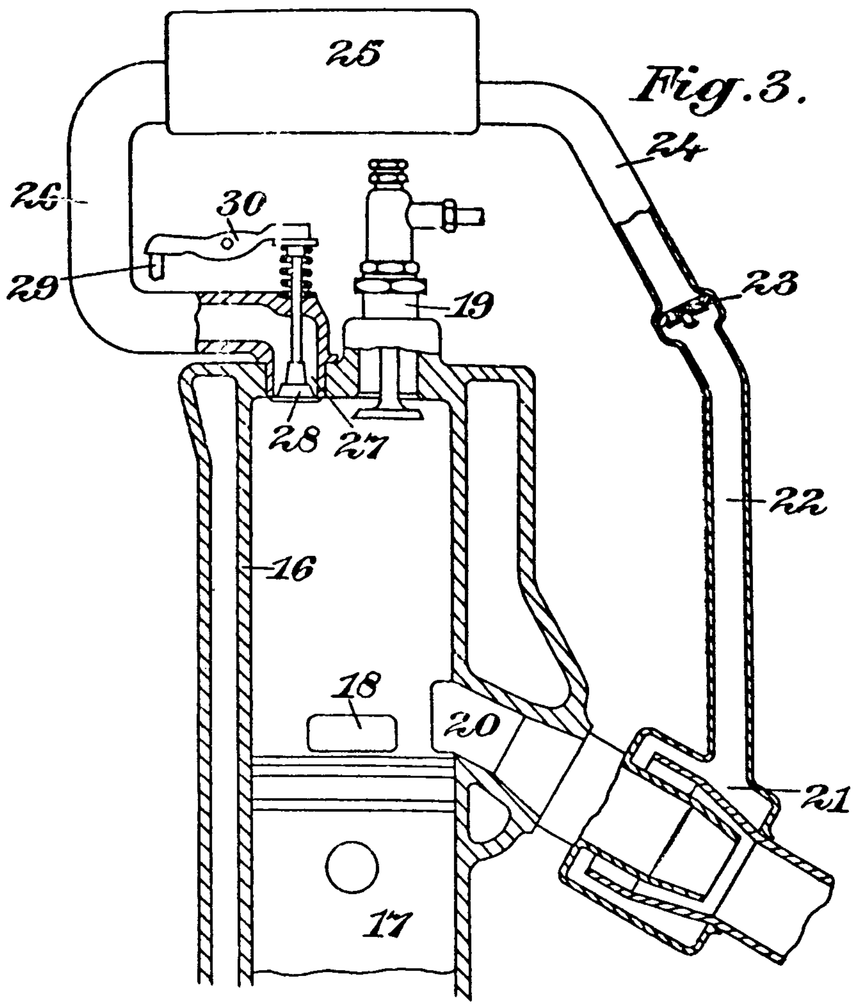

Fig. 3.

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Patented Feb. 14, 1939 2,147,200

UNITED STATES PATENT OFFICE

2,147,200

INTERNAL COMBUSTION ENGINE

Michel Kadenacy, Paris, France

Original application January 23, 1936, Serial No. 60,529, now Patent No. 2,134,920, dated November 1, 1938. Divided and this application June 3, 1937, Serial No. 146,297. In Great Britain August 13, 1935

2 Claims. (Cl. 60—32)

This invention relates to two-stroke cycle internal combustion engines of the kind wherein at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when an adiabatic flow only is involved, and in such a short interval of time that it is discharged as a mass leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder by opening the inlet orifice with the required delay after the opening of the exhaust orifice to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct and that a suction effect is exerted at the inlet orifice as a consequence of the exit of the said mass, and is divided from my co-pending application, Serial No. 60,529 filed 23rd January, 1936, which has issued as Patent No. 2,134,920 on November 1, 1938.

According to the present invention that portion of the depression created by the mass exit of the burnt gases which is not utilized for the purpose of introducing fresh charge into the cylinder is employed for exerting a suction in the engine cylinder or for accumulating a depression in a container which depression may be utilized for example, for exerting a suction in the engine cylinder.

An embodiment of the invention will now be described, simply by way of example, and with reference to the accompanying drawings, in which:

Figure 1 is a curve of pressures and depressions taken in the exhaust duct of an engine of the kind to which the invention relates, during the exhaust period.

Figure 2 shows an example of a suitable arrangement of the communication between the exhaust duct and the pipe to be provided thereon.

Figure 3 shows an arrangement in which the invention is employed for exerting a suction through the cylinder of the engine to which it is applied.

If a record is taken of the pressure variations in the exhaust pipe of an internal combustion engine of the kind to which the invention relates during the exhaust period, a curve similar to that shown in Figure 1 may be obtained, in which EO represents the opening of exhaust, the ordinates represent pressures above and below atmospheric pressure and the abscissae crank angles in degrees.

It should be mentioned that the figure is a formal representation showing the chief characteristics of the curve and that the pressures above and below atmospheric pressure are not shown in scale relationship.

Such a curve may be obtained for example by utilizing a stroboscopic device formed by a ported tube mounted in or on the exhaust duct and rotating with the engine and a stationary but angularly adjustable ported sleeve on this tube having its port connected to a manometer, a pressure or depression impulse being obtained each time the port in the tube and sleeve coincide and a record being taken when a steady reading is given on the manometer, the crank angle at which each reading is taken being determined by the angular adjustment of the sleeve.

The curve shows clearly the two phases of pressure or impulse P, P' that occur during the outflow and return of the gases and the intervening phase of depression D. This curve is characteristic for all internal combustion engines of the kind to which the invention relates but the moments at which the outflow and return of the gases occur will vary.

The period during the operation of the engine at which the aspiration of fluid may be produced in accordance with the invention, will be seen from Figure 1.

In the first place a pressure is registered shortly after the opening of exhaust, as shown by the part P of the curve.

A depression is then formed in the cylinder and a little later a depression D is formed in the exhaust duct and the intensity and magnitude of the volume in which it exists will be proportionate to the kinetic energy contained in the exhaust gases.

At this moment the aspiration of external fluid through the medium of a communication with the interior of the exhaust duct may be obtained.

The return shock P' then follows and destroys this depression phase.

In an engine of the kind to which the invention relates, the fresh charge admitted to the cylinder cannot fill the complete void left in the cylinder and in the exhaust duct by the issuing gases for practical reasons depending upon the position, shape and surface of the admission orifices.

In fact, the applicant has found that if an orifice is provided in the exhaust duct close to the cylinder and is opened to a source of gaseous fluid external to the exhaust duct, the volume of gaseous fluid drawn directly into the exhaust duct through this orifice during the depression phase described above does not in any way impede or reduce the charge admitted directly into the

cylinder through the usual admission orifices, although the volume of gaseous fluid drawn into the exhaust duct may be equal to this charge.

Consequently, by providing a pipe communicating with the interior of the exhaust duct of the internal combustion engine at a point nearer the cylinder than the point from which the burnt gases return, and with a source of fluid external to the exhaust duct, this fluid can be aspirated into the said chamber and thereafter discharged into the exhaust duct, without detriment to the utilization of the depression left in the cylinder by the issuing exhaust gases for the introduction of a fresh charge through the main admission ports.

Suitable distribution means must be provided so that the aspirated air or other gaseous fluid will follow the required direction during the suction and so that it cannot return from the paths it is required to follow.

These means may for example consist of non-return valves or of controlled valves or of means such as those described in British patent specification No. 431,857.

The communication with the interior of the exhaust duct must be suitably arranged to permit a utilization of the depression phase in accordance with the invention.

Figure 2 illustrates one example of an arrangement of the communication with the interior of the exhaust duct.

In this figure the exhaust duct is formed by two portions 10 and 11 connected together by a chamber 12.

The portion 10 of the duct is extended into the said chamber by means of a tubular element 13 opening into the portion 11 and situated in the interior of a tubular element 14 which extends the portion 11 of the duct and stops short of the inner wall of the said chamber 12. The annular space left between the elements 13 and 14 establishes a communication between the interior of the exhaust duct and the chamber 12.

The portion 11 of the duct is adjustably connected with the chamber 12 in order to permit a regulation of the distance between the free end of the element 14 and the internal wall of the chamber situated towards the cylinder.

The diameter of the free end of the element 13 is slightly smaller than that of the duct 11 and the element 14 at this point is slightly flared so as to provide a passage of increasing section between the elements 13 and 14 from the free end of the element 13.

The length of the element 13 may also be regulated by means of the screw connection provided between this element and the duct 10. These adjustments enable the action of the device to be varied so as to vary the intensity of the suction which will be exerted through the outlet 15 of the chamber 12.

In carrying the invention into effect the depression may be employed in order to exert a suction on the cylinder, through an additional outlet other than the main exhaust port, with the object of prolonging the suction in the cylinder, or of intensifying the suction therein, or of sucking residual gases from the cylinder, if desired through the intermediary of a reservoir and suitable distribution means, whereby the depression can be accumulated and employed at chosen moments. An example of such an application is illustrated in Figure 3.

This figure shows an engine cylinder 16 in which moves a piston 17. Air is admitted by atmospheric pressure through the inlet 18 and fuel is introduced by the injector 19. Exhaust takes place through the duct 20.

Upon the exhaust duct close to the cylinder is provided an intake 21, which by way of example is shown similar to that illustrated in Figure 2, communicating with a first chamber 22.

This chamber 22 is provided at its other end with a suction valve 23 or the like, which may be controlled or otherwise, and which allows the passage of fluid only in the direction towards the exhaust duct. This valve 23 is followed by a duct 24 leading to a reservoir 25 connected by a duct 26 to an additional outlet 27 provided on the cylinder and controlled by a valve 28 actuated by a push rod 29 and rocker arm 30.

With such an arrangement a depression will be detained in the reservoir 25 which may be utilized at a convenient time by a suitably timed operation of the valve 28, in order to draw residual gases from the cylinder or to assist the entry of the charge to the cylinder through the inlet port 18.

In this example the suction valve 23 opens automatically each time a depression is left in the exhaust duct but it is obvious that the valve 28 should not open before the exhaust gases have left the cylinder and that it should not remain open after the inlet port 18 has closed.

In the example described with reference to Figure 3, the charge is introduced into the cylinder by atmospheric pressure but this does not exclude the use of means for maintaining or augmenting the pressure of the supply.

Any suitable devices may be provided in the exhaust duct in order to prevent a return wave of the exhaust gases from re-entering the cylinder, such for example as the means described in the applicant's British patent specification No. 431,857.

The valves employed for the suction of the fluid in an arrangement according to the invention may be simple or multiple. These valves may be arranged so as to provide a passage of large area and they may have any shape provided they respond to rapid suctions.

I claim:—

1. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the exhaust conduit providing a permanent free passage for the burnt gases to the limit of outward travel of said gases, and providing a passage for the gases during their outward motion as a mass having no cross section of substantially greater area than any cross section thereof further from the cylinder, an intake on the exhaust conduit at a point situated nearer the cylinder than the limit of outward travel of the burnt gases, a supplementary outlet on the cylinder, means for so controlling said outlet as to ensure that it will be opened after the said issuance of the burnt gases through the exhaust orifice, a duct connecting the said intake to the supplementary

outlet, controlling means in said duct, said controlling means opening towards said intake, whereby the depression left behind the mass of burnt gases when it has passed beyond said intake causes a suction effect to be exerted in said connecting duct whereby any residual gases may be drawn out of the cylinder upon the opening of the supplementary outlet or the suction in the cylinder during charging may be prolonged or intensified.

2. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the exhaust conduit providing a permanent free passage for the burnt gases to the limit of outward travel of said gases, and providing a passage for the gases during their outward motion as a mass having no cross section of substantially greater area than any cross section thereof further from the cylinder, an intake on the exhaust conduit at a point situated nearer the cylinder than the limit of outward travel of the burnt gases, a supplementary outlet on the cylinder, means for so controlling this outlet as to ensure that it will be opened after the said issuance of the burnt gases through the exhaust orifice, a duct connecting the said intake to the supplementary outlet, a reservoir in the said connecting duct, controlling means in the conduit between the reservoir and the said intake, the said controlling means opening towards the said intake whereby a depression may be stored in said reservoir for utilization in prolonging or intensifying the suction in the cylinder during charging or in sucking residual gases from the cylinder through the supplementary outlet.

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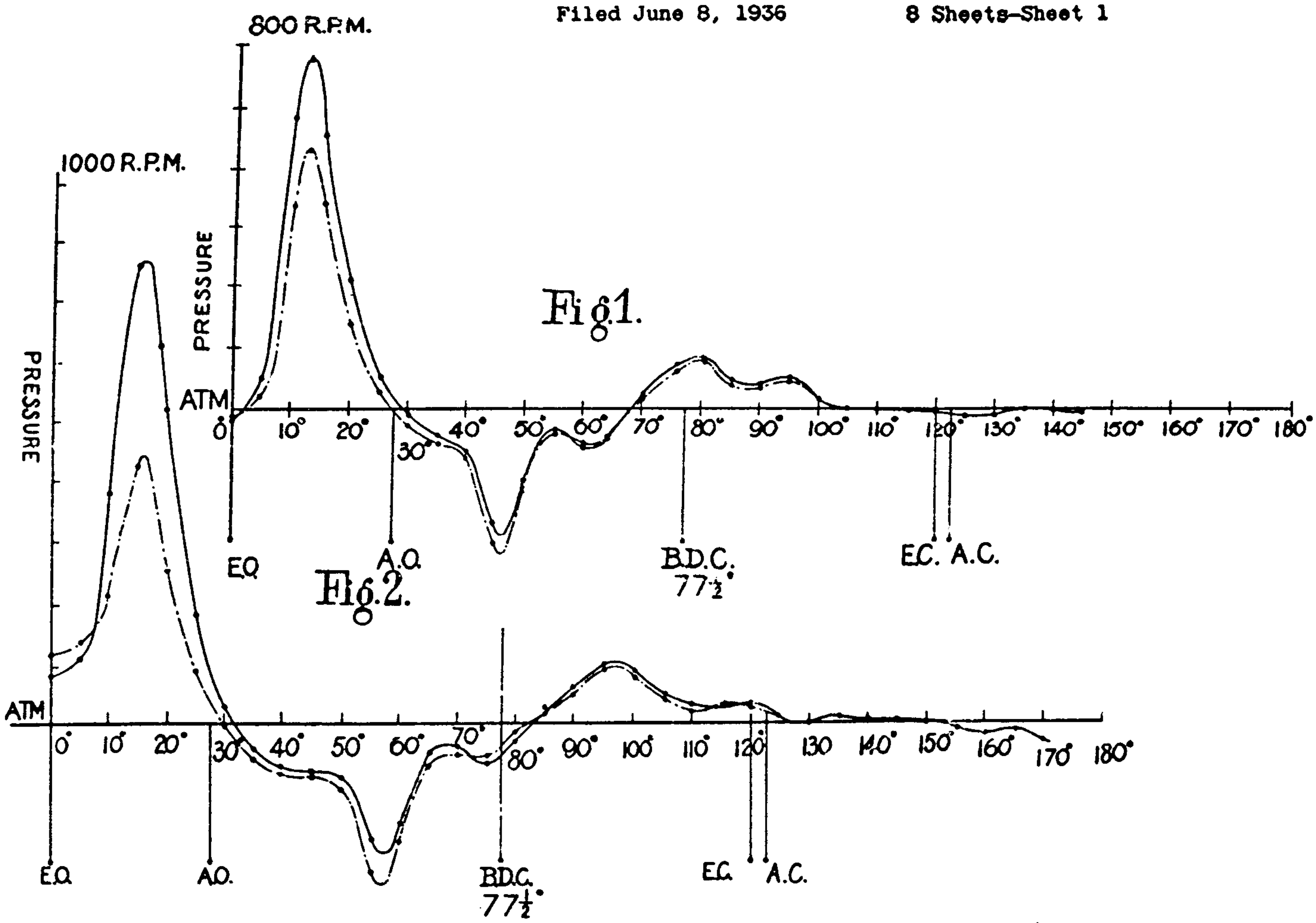
Sept. 20, 1938. M. KADENACY 2,130,721

TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed June 8, 1936 8 Sheets-Sheet 1



Inventor M. Kadenacy

By: Glascock, Downing & Seebold Attys.

Sept. 20, 1938. M. KADENACY 2,130,721
TWO-STROKE INTERNAL COMBUSTION ENGINE
Filed June 8, 1936 8 Sheets-Sheet 2

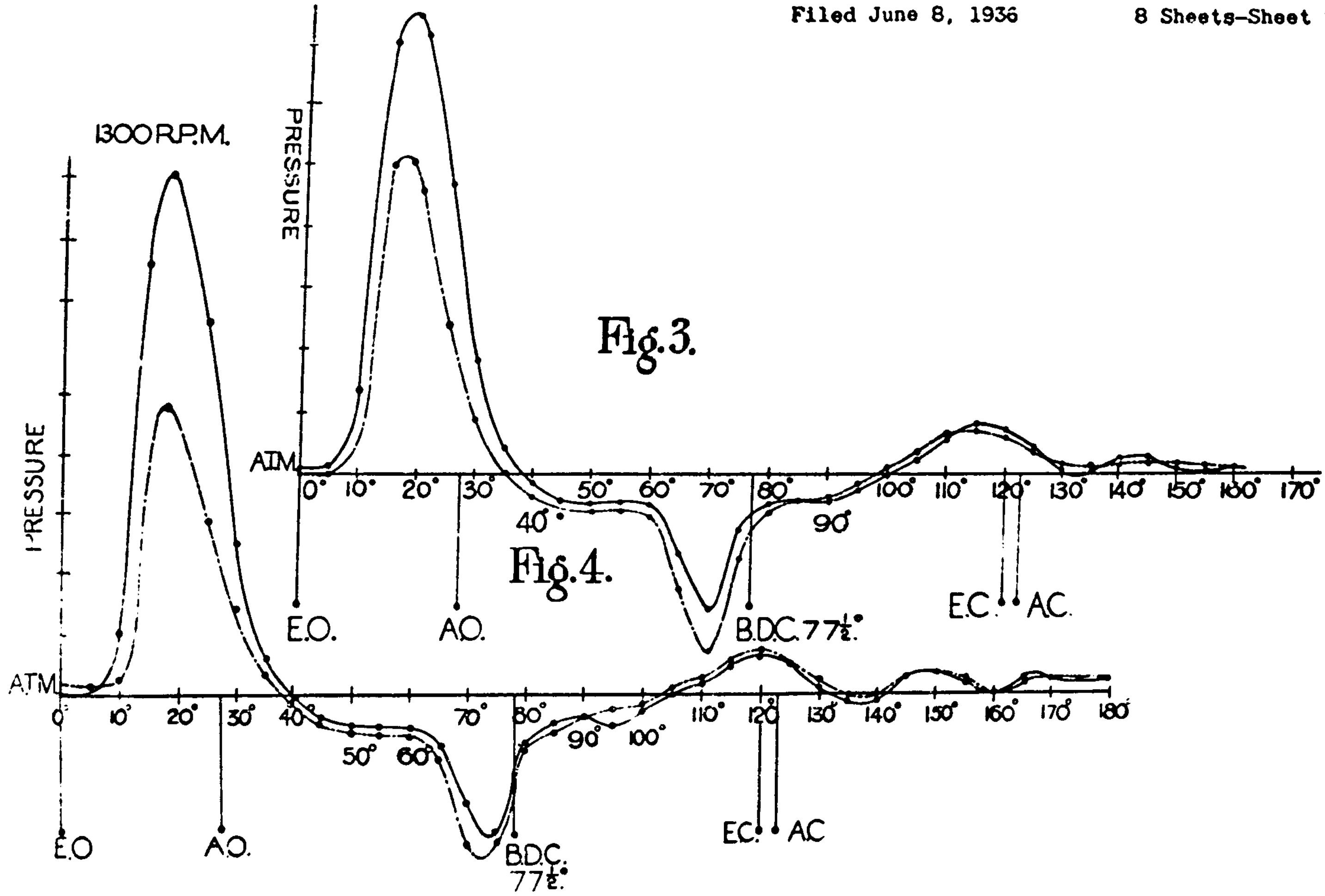


Fig. 3.

Fig. 4.

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8 Sheets-Sheet 3

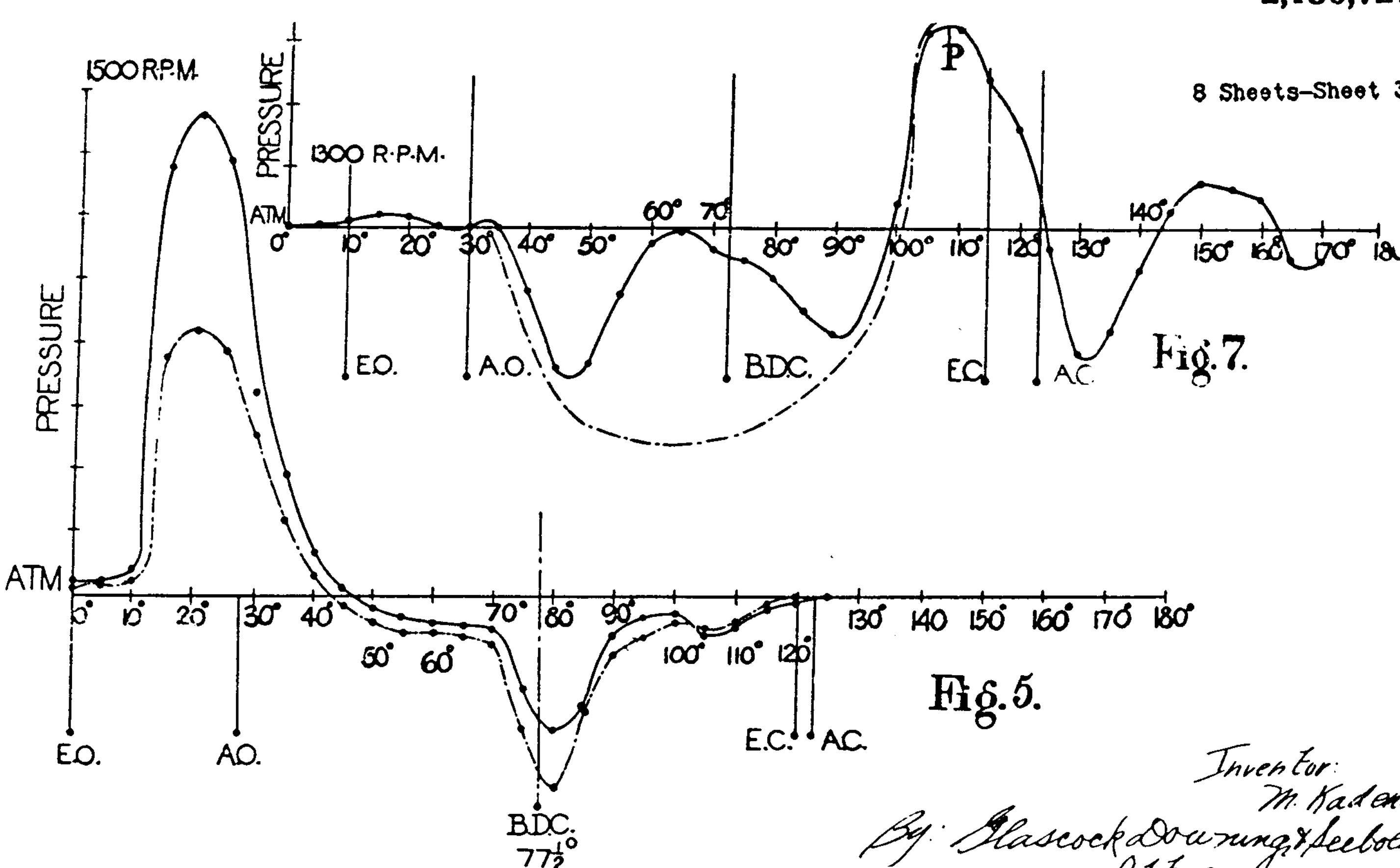

Fig. 7.

Fig. 5.

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TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed June 8, 1936 8 Sheets-Sheet 4

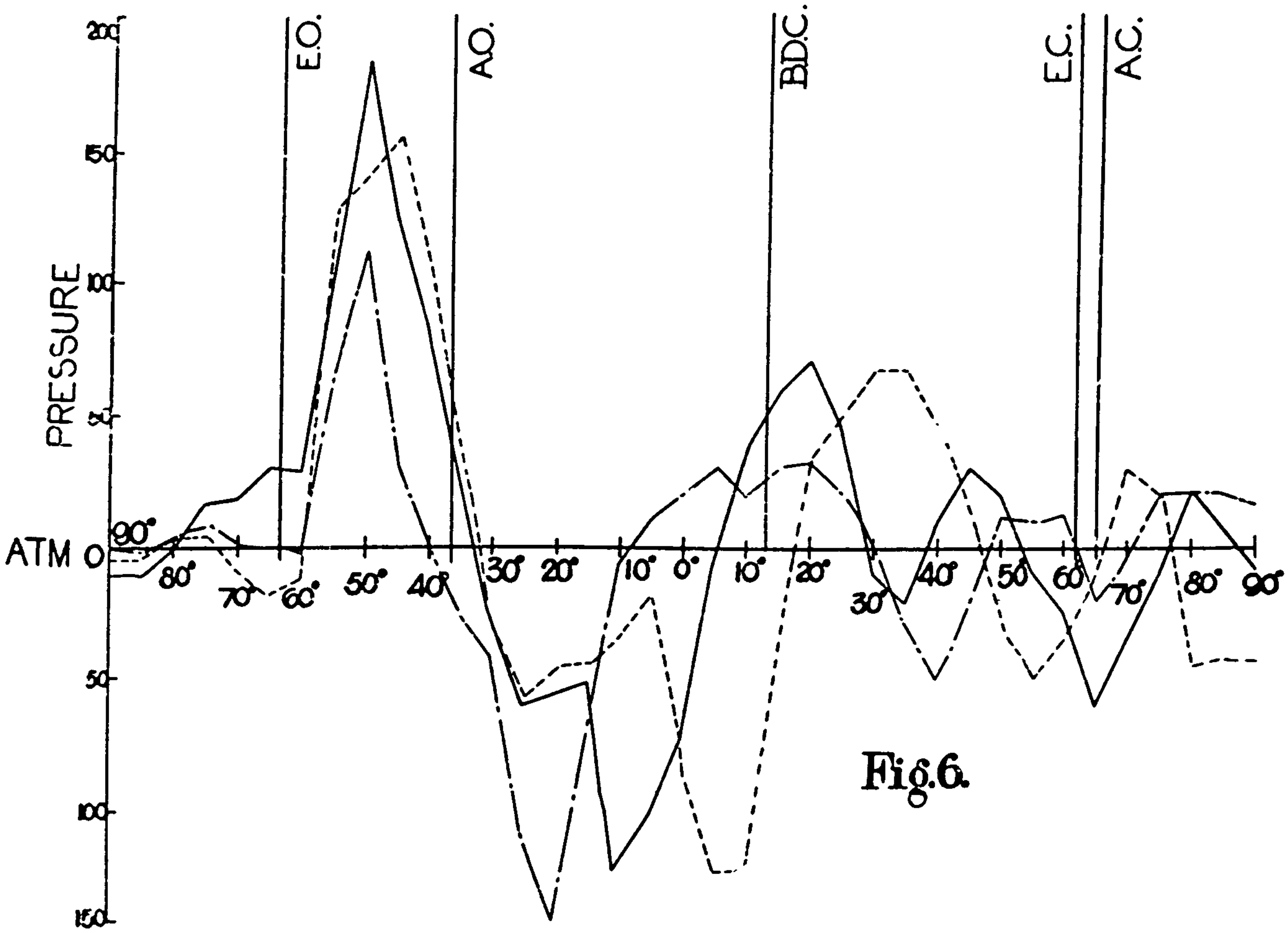

Fig.6.

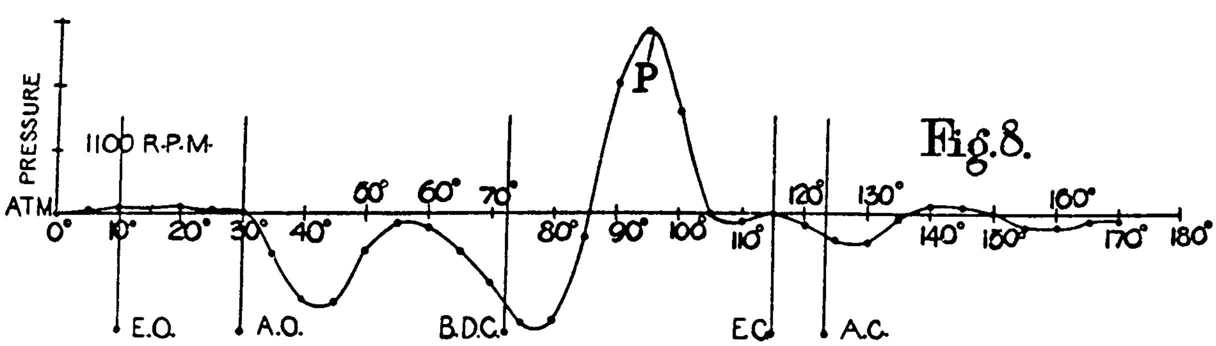

Fig.8.

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8 Sheets-Sheet 5

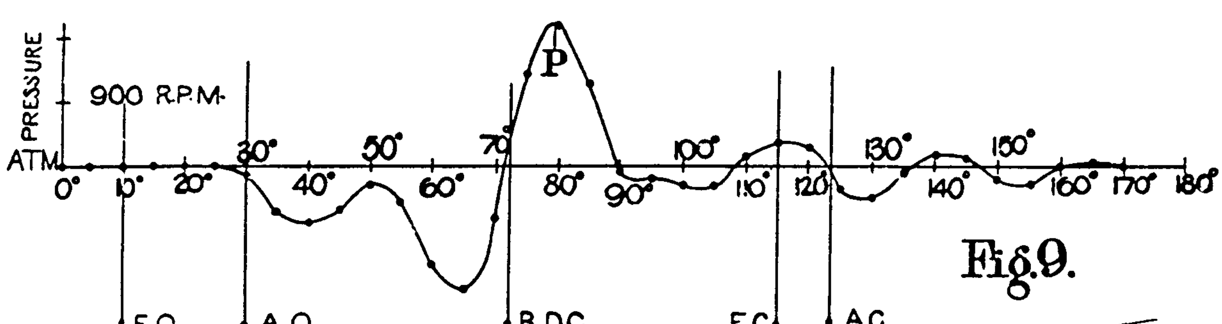

Fig.9.

Inventor: M. Kadenacy
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Sept. 20, 1938. M. KADENACY 2,130,721

TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed June 8, 1936 8 Sheets-Sheet 6

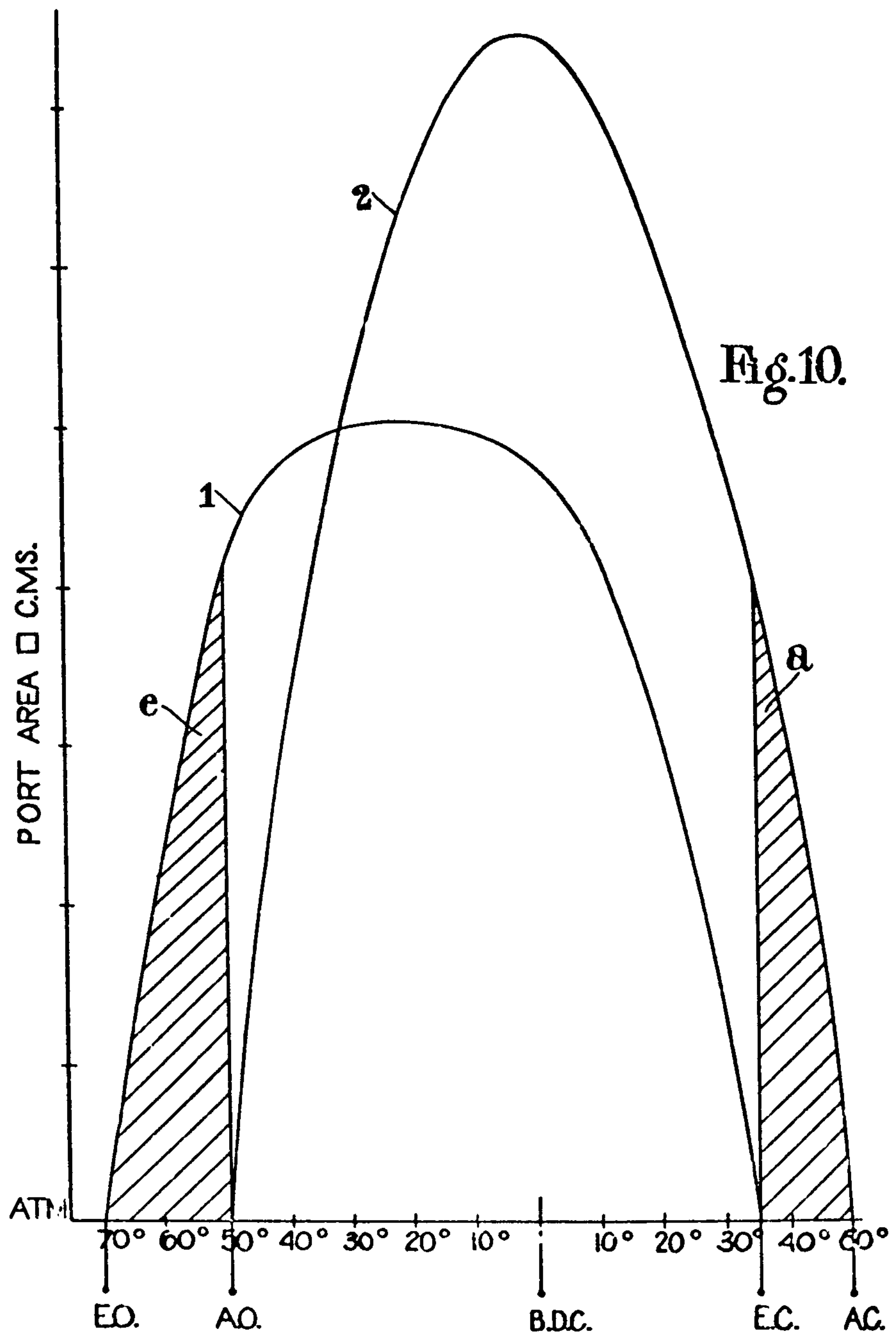

Fig.10.

Inventor: M. Kadenacy
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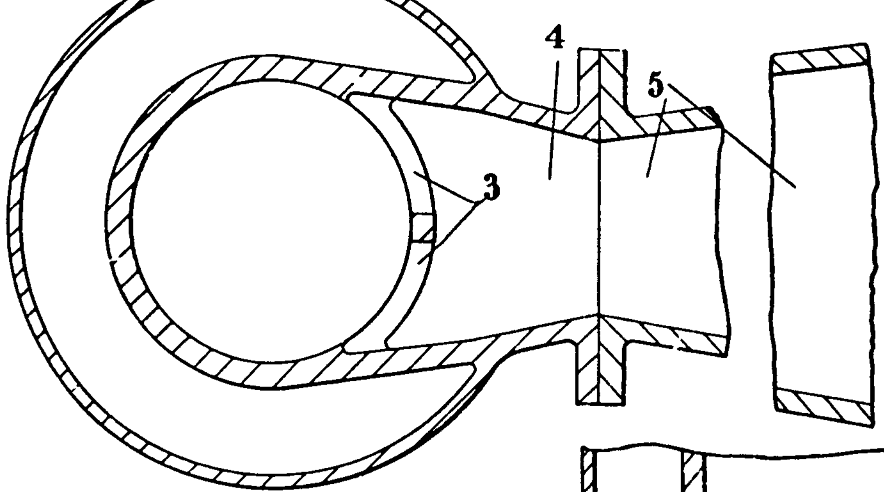
Sept. 20, 1938. M. KADENACY 2,130,721

TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed June 8, 1936 8 Sheets-Sheet 7

Fig.11.



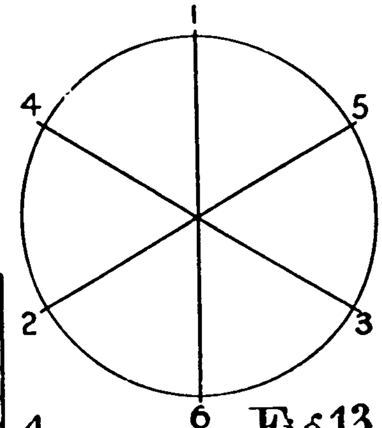


Fig.13.

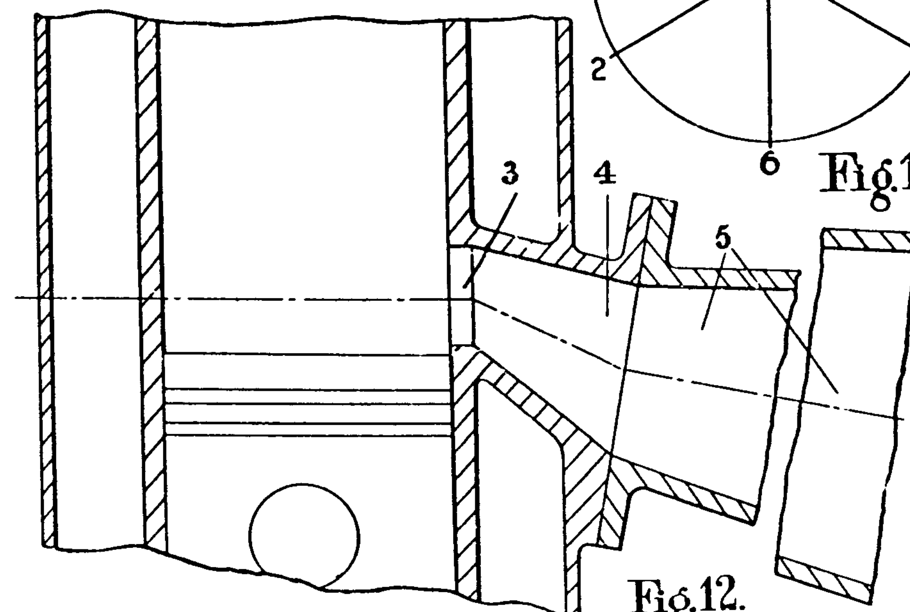


Fig.12.

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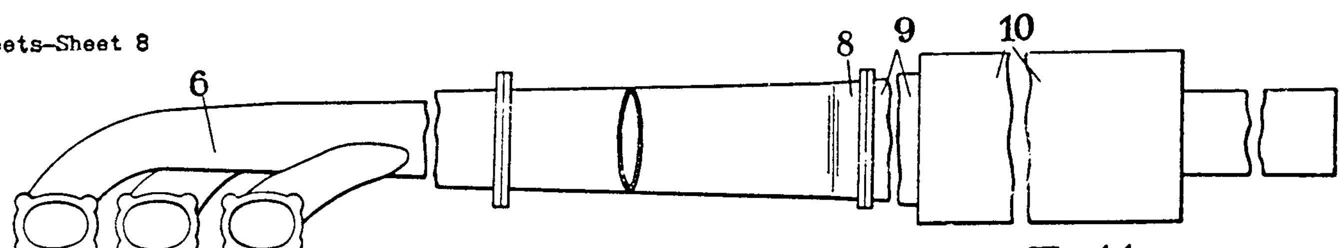

Fig.14.

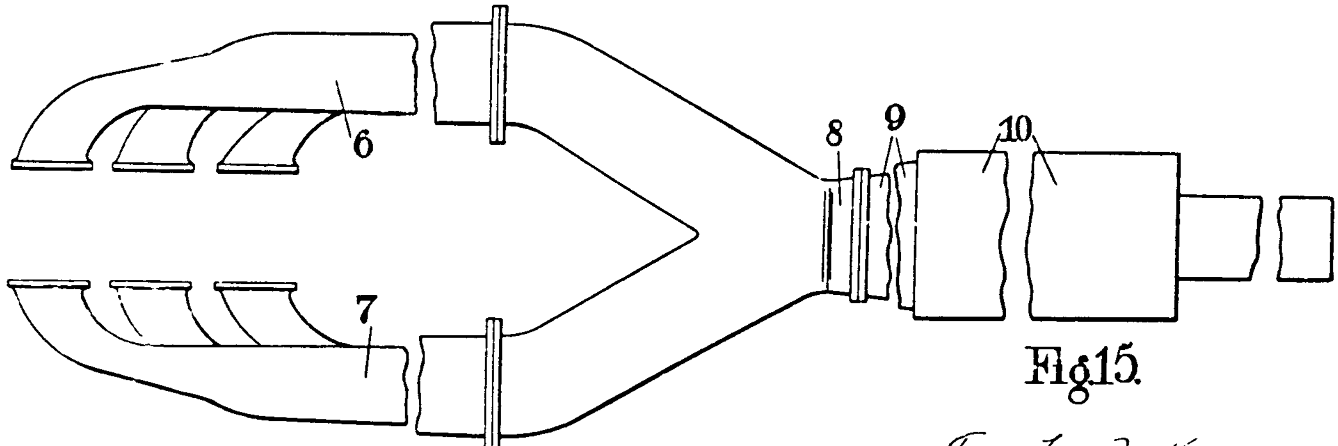

Fig.15.

Inventor: M. Kadenacy

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Patented Sept. 20, 1938 2,130,721

UNITED STATES PATENT OFFICE

2,130,721

TWO-STROKE INTERNAL COMBUSTION ENGINE

Michel Kadenacy, Paris, France, assignor of one-half to Armstrong Whitworth Securities Company Limited, London, England

Application June 8, 1936, Serial No. 84,182
In Great Britain January 11, 1936

4 Claims. (Cl. 123—65)

This invention relates to a method of constructing two-stroke cycle internal combustion engines of the kind wherein at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when an adiabatic flow only is involved, and in such a short interval of time that it is discharged as a mass leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder.

The object of the invention is to provide a method of constructing an improved engine of the above kind.

The invention comprises selecting a crank angle for the charging period, establishing the area of the inlet orifice and the rate of opening of this orifice so that a working charge can enter by atmospheric pressure during the charging period as a consequence of the mass exit of the burnt gases from the cylinder, selecting a moment for opening the exhaust orifice, and making the area of the exhaust orifice and the rate of opening of this orifice such that the area of exhaust orifice opened before the opening of inlet ensures that the mass exit of the burnt gases occurs before the opening of inlet.

The present inventor has already indicated in prior specifications that the behavior of the gases upon and after their discharge from the cylinder is such as to lead to the belief that the burnt gases while still in the cylinder form a body having properties similar to those of a resilient body, and which upon the opening of an exhaust orifice seeks to project itself as a mass from the cylinder. He has observed that when the exhaust orifice opens there is first a period of delay during which the gases do not emerge from the cylinder, and that after this delay has elapsed the burnt gases issue from the cylinder at a speed greatly in excess of the speed obtaining for an adiabatic flow and as a mass the motion of which is governed by the laws of reflection and rebound.

This high velocity will hereinafter be referred to as the ballistic speed of the burnt gases, and the force causing this high velocity will be referred to as the ballistic energy of the burnt gases.

In engines of the type to which the present invention relates the whole or at least a substantial portion of the burnt gases leaves the cylinder ballistically as a mass. If all the exhaust gases leave ballistically as a mass the cylinder will be left substantially void of burnt gases. If only a substantial portion of the exhaust gases leaves ballistically as a mass the burnt gases in the cylinder will be left in a highly rarefied condition.

The void or depression left behind the mass of burnt gases exists not only in the cylinder, but also in the space into which the burnt gases issue, e. g., the exhaust duct or system, and the length of time this void or depression is maintained is determined by a return of the rear zone of the gaseous mass on its return rebound towards the cylinder.

It will be understood that if a portion of the ballistic energy of the burnt gases is destroyed during their mass exit from the cylinder the volume of the void left behind the gases and the duration of time for which this void is maintained will be correspondingly reduced.

Another object of the invention is therefore to provide a two-stroke internal combustion engine as set forth above, in which the burnt gases retain the maximum proportion of their ballistic energy when they leave the cylinder, and in which a maximum utilization is made of the void left by the burnt gases upon their mass exit from the cylinder.

With this object in view, the invention further consists in selecting a crank angle for the charging period, in establishing a maximum area of the inlet orifices and a maximum rate of opening of these orifices, within mechanical limits, establishing exhaust orifices having a maximum area within mechanical limits and establishing the period between the opening of the inlet orifices and opening of the exhaust orifices such that the inlet orifices will open when the burnt gases commence to leave the cylinder as a mass and arranging for the area of exhaust orifices which is opened prior to the opening of the inlet orifices to be opened at a maximum rate within mechanical limits, whereby the maximum charge may be introduced into the cylinder in the time available.

The invention also consists in providing means for combating any objectionable influence of a return of the burnt gases or of a prolonged suction on the charge contained in the cylinder.

Further features of the invention will appear from the following description in which reference will be made to the attached drawings.

In the drawings:

Figures 1 to 5 are records of pressures taken during the exhaust and inlet periods in the exhaust pipe of an engine according to the invention, close to the cylinder, the figures corresponding respectively to speeds of 800, 1000, 1200, 1300 and 1500 R. P. M.

Figure 6 shows three records of pressures taken on the exhaust duct of another engine, at one speed but with different exhaust systems on the engine.

Figures 7, 8 and 9 are records of pressures taken on the inlet duct of an engine during the exhaust and admission periods.

Figure 10 is an exhaust and inlet port area diagram.

Figures 11 and 12 show by way of example a cross section through a cylinder of an internal combustion engine showing the exhaust ports and the passages leading from these ports into an exhaust pipe.

Figure 13 is a timing diagram relating to a six-cylinder engine.

Figures 14 and 15 show an arrangement of exhaust duct and manifolds in accordance with the invention for a six-cylinder engine.

The records shown in Figures 1–9 were all obtained by means of the device described in application Serial No. 82,958 filed June 1, 1936.

In these figures the ordinates represent pressures above and below atmospheric pressure and the abscissae represent crank angles. EO, EC, AO, AC, indicate respectively exhaust opening and closing and inlet opening and closing, while bottom dead centre is indicated at BDC.

The curves shown in Figures 1 to 5 are differential pressure curves which also enable the velocity and direction of movement of the gases to be observed.

Each of these figures shows two similar curves, one in full lines and one in dotted lines. Where the full line curve lies above the dotted line curve, the direction of motion is away from the cylinder.

The difference between the ordinates of the two curves at any point is also an indication of the velocity of the gases at any moment.

By referring to any one of Figures 1 to 5, it will be seen that when the exhaust port opens there is first a delay during which substantially no change in pressure occurs in the exhaust pipe and that thereafter the pressure rises abruptly to a peak, then falls abruptly to a pressure below atmospheric pressure and that the pressure subsequently rises again and destroys the depression.

If these changes in pressure are considered in conjunction with the direction of motion of the gases, it will be seen that during the period of delay, a substantially constant and very small pressure difference exists in the exhaust pipe. Thereafter the pressure difference increases very rapidly to a maximum pressure difference which remains substantially constant for a little distance on either side of the pressure peak of the curve. This can be taken as representing the passage of the mass of burnt gases at high velocity past the indicating device.

Thereafter the movement continues to be in the direction away from the cylinder, but by this time the mass of burnt gases has passed the indicator and is at some point further along the exhaust pipe and the pressure difference indicates the state of the medium which is at that moment passing the indicator.

It will then be seen that at a later point in the crank angle, the two curves cross indicating a reversal in the direction of motion and that this reversal is followed by the return impact of the burnt gases.

The curves shown in Figures 1 to 5 were obtained from an engine constructed in accordance with the invention. It will be seen that inlet opens substantially when the tail end of the mass of burnt gases has left or is about to leave the cylinder, and it will also be seen from the curves that the entering charge arrests the rapid fall in pressure and partly fills the void left behind the mass of burnt gases. Subsequently the pressure again falls in the exhaust pipe before the reversal in direction occurs, showing that in this example the entering charge cannot fill completely the void left by the burnt gases in the exhaust system.

If the inlet orifice instead of being opened in the manner indicated above had remained closed when the burnt gases left the cylinder, the curves would have shown a fall in pressure to a very high degree of vacuum and an equally abrupt return to a high pressure, since there would be nothing in the cylinder to oppose or attenuate this return, and it should be understood from the above that the form of the curve is modified by the fact that the entering charge follows the issuing burnt gases, although the characteristics of the curve remain unaltered.

From the curves it can be seen that the rise in pressure to the peak and the subsequent drop in pressure are extremely abrupt and cannot be compared with an adiabatic expansion and that the total exhaust occupies an interval of time which is shorter than that which would be required for the burnt gases to expand adiabatically down to atmospheric pressure, and it can be concluded that the true and instantaneous speed of mass exit from the cylinder is much higher than the mean overall speed of exit and is higher than the speed of sound.

In this connection it should be noted that the indicator records changes of pressure in the exhaust pipe close to the cylinder. If the accompanying changes in pressure in the cylinder are considered it will be understood that the abrupt rise in pressure to the peak of the curve corresponds to an equally abrupt drop in pressure in the cylinder behind the gases.

Further, it can be shown by multiple observations that after the exhaust the cylinder remains under a depression and it will be seen from the curves that this depression is prolonged for a certain duration of time, after which the return impact occurs.

From the above observations it can be concluded that the gases before their exit from the cylinder have an initial velocity proper and that adiabatic expansion only intervenes as a secondary phenomenon.

The above and other observations have led the inventor to conclude that the mean speed of ballistic exit of the burnt gases from the cylinder is of the order of 1400 to 1800 metres per second, taking into account the fact that this exit does not occupy the total time interval elapsing between exhaust and inlet opening, in view of the initial delay that must exist and the tolerance that must be allowed in a variable speed engine.

For the purpose of this description the total exhaust of the burnt gases is considered to be the period elapsing from the moment of opening the exhaust orifice to the moment when the tail end of the issuing mass of burnt gases leaves the cylinder.

The delay that occurs at the commencement of this period before the burnt gases engage in the exhaust system may be explained by a consideration of the viscosity and dynamic inertia of the burnt gases and the static inertia of the gaseous mass external to the exhaust orifice.

The gaseous medium external to the exhaust orifice possesses a static inertia that has to be overcome by the burnt gases before the latter can issue from the cylinder.

But when the exhaust orifice first commences to open and only a narrow slit is open, the mass of burnt gases cannot on account of its viscosity emerge from the cylinder and engage fully with the gaseous medium external to the exhaust orifice, and so is unable to exert its full force on this external gaseous medium.

When such a sufficient area of exhaust orifice has opened, the burnt gases may not at this moment be directed towards the exhaust orifice, in other words, there may be a delay due to the dynamic inertia of the burnt gases.

The true moment of commencement of exhaust will be when a sufficient area of exhaust orifice has opened for the burnt gases to emerge bodily from the cylinder and the said gases issue from the cylinder as a mass and overcome the resistance of the external gaseous medium.

The above effects explain the period of delay that occurs at the commencement of the curves shown in Figures 1 to 5 before the abrupt rise in the pressure occurs.

When leaving the cylinder the mass of burnt gases forms a column in passing through the exhaust orifices, and the time occupied by the exit of the burnt gases is affected by the length of this column, that is by the area of the exhaust orifice and by the necessity of deforming the viscous mass of burnt gases during their passage through this orifice.

If the angle between exhaust opening and inlet opening has been determined in advance it will be understood that a sufficient area of exhaust orifice must open before inlet opens to ensure that the mass exit of the burnt gases from the cylinder occurs within this angle, and this may be arrived at from a consideration of the mean velocity of total exit of the burnt gases referred to above and of the mean area of the exhaust orifice opened in the angle in question and the fact that the volume of burnt gases must be discharged in an interval t of time which is shorter than that required for the adiabatic expansion to manifest itself as a dominating factor.

For the purpose of making such a calculation the time interval t within which the total exhaust must be effected may be taken as being of the order of 1/300 sec. It may of course with advantage be smaller than this amount.

The inventor has found that calculations of sufficient accuracy to ensure practical results may be made by assuming that the cylinder volume of burnt gases is discharged, without expansion, at a hypothetical mean speed. This hypothetical mean speed of discharge will vary according to the fuel employed, the mixture and the conditions of combustion, among other factors. For fuel oil with good combustion a hypothetical mean speed of 450 metres per second may be assumed, although this hypothetical speed may be as low as 300 metres per second or as high as 700 metres per second.

By way of example if

N is the number of revolutions per second of the engine

A is the area of exhaust orifice opened before inlet opens in cm^2

K is a constant depending upon the form of exhaust orifice and the area opened per unit movement of the piston or crankshaft

KA being the means effective area of exhaust orifice opened before inlet opens

W is the volume of the cylinder in cm^3

v is the mean velocity of mass exit of the burnt gases in metres per second

a is the angle of exhaust lead

Then the length of the column formed by the passage of the mass of burnt gases through the exhaust orifice will be

$$\frac{W}{100KA}\text{ metres}$$

The time interval occupied by this mass exit will be

$$\frac{W}{100KA,}\text{ secs.}$$

The time elapsing between exhaust opening and inlet opening will be

$$\frac{a}{360N}\text{ secs.}$$

so that the following relationship should exist

$$\frac{a}{360N}=\frac{W}{100KA,}=t$$

But it will be seen that in order to reduce the period of total exit of the burnt gases to a minimum the area of the exhaust orifice should be made as large as possible, and the area of exhaust orifice which is opened prior to the opening of the inlet orifice should be opened at a maximum rate, all within mechanical limits. In this case it will be more simple to open the full area of the exhaust orifice at a maximum rate but it will be understood that the rate of opening the remaining area of the exhaust orifice after the gases have left the cylinder is not of importance.

In this way if the angle available for evacuating and charging the cylinder has been determined in advance, the maximum portion of this angle will be available for the purpose of charging.

The choice of such an angle will depend upon the construction of the engine and will be determined in accordance with well-known principles in order to give the maximum useful cylinder volume.

By experiment it has been found by the inventor that the passage to the atmosphere can be opened when the outgoing gases have left the place at which the admission orifice is situated. For example in an opposed piston engine wherein the inlet and exhaust orifices are at opposite ends of the cylinder, the admission orifices can be opened when the exhaust gases are still in the bottom of the cylinder, near the place at which the exhaust orifices are situated; this is explicable by the viscosity of the gases and by the velocity with which they are leaving.

From the foregoing, and as described in application Serial No. 738,014 filed August 1st, 1934, it will be seen that if the inlet is opened at the moment when the issuance of the burnt gases as a mass is in full progress and causes a suction effect to be exerted in the cylinder a fresh charge may be admitted by the action of the ambient atmospheric pressure. More precisely, inlet should open when the rear face of the issuing mass of burnt gases leaves the exact position of the inlet orifices.

If the opening of inlet is delayed, the time available for charging is reduced, and the return impact occurs more quickly and with greater violence. The curves show that the latest moment

at which such an opening could be effected is immediately before the reversal in direction of motion occurs.

In order to obtain the maximum result inlet should open without delay and with the greatest area opened per unit of movement of the piston. But in practice a fairly large tolerance is permissible from the constructional point of view in order to obtain a highly satisfactory operation of the engine.

It is therefore possible to choose a moment for opening the inlet orifice or orifices which is situated relatively close to the moment of exhaust of the burnt gases and which remains suitable for charging the engine over a wide range of speeds.

In Figures 1 to 5 it will be seen that the timing of inlet opening remains unaltered and that over the range of speeds represented by the curves, this opening of inlet is always relatively close to the moment of exit of the burnt gases.

For example in Figures 1 to 5 inlet is timed to open 28½° after exhaust and this timing remains suitable over at least the speed range represented by the figures. The volume of the engine cylinder is 700 ccs. and the area of exhaust orifice opened before inlet opens is 13.8 sq. cms. If the rate of opening of the exhaust orifices is increased the inlet may be opened 20° after exhaust when 13.2 sq. cms. of exhaust orifice have opened. The maximum rate of opening that is mechanically possible in this engine will permit inlet to be opened 18° after exhaust when 13.6 sq. cms. of exhaust orifice have opened. In all three cases the total area of exhaust orifice is larger than that open at the moment of inlet opening, and approaches the maximum area that can be allowed.

In order that the cylinder may be filled with a fresh charge, the inlet orifices must have a sufficiently large area and be open for a sufficiently long time in order that a fresh charge can enter under atmospheric pressure into the cylinder and into a part of the exhaust duct.

If the angle available for charging has been determined, the area of the inlet orifice that must be opened to ensure the entry of a sufficient charge may easily be determined from a consideration of the known mean speed of expansion of air from atmospheric pressure into a void and of the volume of the void to be filled.

For example a calculation may be based on the requirement of introducing say 1.5 cylinder volumes of air into the cylinder and exhaust duct, and a mean speed of entry of the charge of 50 to 60 metres per second may be assumed. This value will be a conservative one and the mean speed of entry for practical purposes may be as high as 100 metres per second.

It will however be preferable and also more simple in practice if the inlet orifice is made as large as possible and the full area of the inlet orifice is opened as rapidly as possible so that this full area will be available for charging for the longest possible portion of the charging period.

The curves show clearly the time necessary and available for the introduction of the charge from which a suitable admission arrangement can be arrived at and how the area and time of opening of the inlet orifice may be established in relation with the characteristics of the exhaust system.

Figure 10 shows a port area diagram for a cylinder of 1.5 litres capacity established in conformity with the above indications. The curve 1 relates to the opening of the exhaust orifice and the curve 2 to the opening of the inlet orifice. The ordinates of the curves represent port areas and the abscissae crank angles.

The portions of the curves which are of great importance are as follows:

(1) The portion *e* of the exhaust curve up to the moment when the inlet orifice opens. This area must be sufficient for the purpose set forth and preferably the slope of the curve should be as steep as possible.

(2) The portions of the curves 1 and 2 that overlap. During this angle both inlet and exhaust are open, and air is passing through the cylinder and the exhaust duct. A variation in this angle will vary the amount of cooling obtained by the air that passes through the cylinder.

(3) The portion *a* of the admission curve, after the exhaust has closed and when the pressure in the cylinder is restored to the ambient pressure. The velocity of the incoming charge will during this period add its action to that due to a difference in pressures.

Now the charge introduced must be retained in the cylinder without being forced out or sucked out by the movements of the exhaust gases which either continue in the direction leading from the cylinder or are transformed into a return impact by the rebound which occurs either in the exhaust ducts or in the open atmosphere.

The curves shown in Figures 7, 8 and 9 indicate the manner in which the reversal in direction of the burnt gases is transmitted through the inlet. In Figure 7 the full and dotted line curves serve to show, as in Figures 1 to 5, the moment when this reversal in direction appears at the inlet.

From these three figures it will be seen that at low speeds the return impact of the gases is liable to be objectionable and that at high speeds the inlet may close before the return occurs.

The return impacts force out and foul the charge; an unnecessarily prolonged suction which follows the admission reduces the charge by placing it under a depression. These two objectionable factors come from outside the cylinder and have a repercussion upon its contents. They must either be suppressed or attenuated, or retarded or separated from the cylinder, so that their influence cannot affect the contents of the cylinder itself. This may be obtained by the timing of the engine itself, by the timing of the exhaust proper, by an arrangement of the exhaust system in form and in volume or by corresponding arrangements in the exhaust system.

For example a suitable exhaust closure may be established as described for example in application Serial No. 84,184 filed June 8, 1936, or in application Serial No. 83,120 filed June 2, 1936, whereby the objectionable factors in question may be separated from the cylinder.

Or the means described in application Serial No. 738,016 filed August 1 1934 may be utilized in order to prevent the return impact from reentering the cylinder.

The introduction of a supplementary compressed charge at the end of the admission period as described in application Serial No. 745,814 filed September 27, 1934 or the injection of air into the exhaust duct at a suitable moment as described in application Serial No. 46,804 filed October 24, 1935 will oppose and retard the return impact.

The use of means such as those described in application Serial No. 38,826 filed August 31, 1935 and in application Serial No. 82,959 filed June

1, 1936 and those described in application Serial No. 46,805 filed October 25, 1935, will permit the said objectionable actions to be opposed and attenuated.

These means are simply indicated by way of example and any means which ensure that the charge will be retained in the cylinder may be employed.

The inventor has found by experiment that the absolute velocity of exit of the burnt gases from the cylinder may be retarded or accelerated according to the nature of the space which the gases enter when they leave the cylinder.

For example exhaust pipes which are of too large dimensions in cross sectional area retard the speed of exit and bring the return impact nearer. Tubes which are too small in cross sectional area retard the exit by compressing the column and deforming the gaseous body; the increase in density of the burnt gases as a consequence of this compression causes the column of exhaust gases to lose momentum, due to the consequently increased friction. In all these cases it can be understood that the actions referred to are those which produce a negative acceleration on the issuing gases. Tubes of suitable dimensions maintain the speed of the gases, and the total exit of the gases from the cylinder occurs more rapidly.

The exit of the exhaust gases directly into the atmosphere, that is, when no exhaust system is present, occurs with a high loss in velocity; the return impact by rebound follows immediately and the duration of time for which the cylinder remains void is shorter than in all the other cases.

A further object of the invention is to indicate the requirements that must be fulfilled by an exhaust pipe for an engine constructed in accordance with the invention, in order to ensure that the issuing mass of burnt gases will be subjected to a minimum deceleration during its outward motion.

As stated above when the exhaust gases leave the engine cylinder consequent upon the opening of the exhaust ports, they tend to form a column, the length of which will be dependent upon the area of the exhaust orifices open during the mass exit of the gases.

At this moment the issuing mass of burnt gases possesses a very high velocity and conform to the laws of reflection.

Consequently upon issuing from the cylinder it should encounter in the exhaust passages and in the exhaust pipe no surfaces capable of reflecting it back into the cylinder or of impeding its motion away from the cylinder.

Further, the energy contained in the exhaust gases is capable of displacing a proportionate mass of the resisting gaseous medium external to the cylinder. If this resisting medium presents a large surface to the issuing mass of burnt gases, the latter will be deformed and flattened, and the issuing column will be of shorter length, its negative acceleration will be greater, and its return to the cylinder will be more rapid.

Exhaust pipes of too large a diameter relative to the exhaust ports will cause this to occur, and the extreme case will be encountered, if the exhaust gases are allowed to issue directly into the open atmosphere.

On the other hand, if the exhaust pipes are of too small a diameter, the resistance to deformation of the issuing mass, which may possess a very high viscosity again exerts too great a decelerating motion on the issuing mass. It retards the complete evacuation of the burnt gases and causes the return to occur more rapidly.

In both cases therefore the time available for effecting the admission is reduced.

According to the invention the walls of the passages and ducts through which the burnt gases pass upon leaving the exhaust orifices are so formed that they always tend to guide and reflect the burnt gases away from the cylinder in the direction of exhaust and the section of passage for the burnt gases through these passages and ducts is made such that the burnt gases upon leaving the exhaust orifices do not thereafter encounter any sudden and considerable increase or decrease in cross section capable of causing an increase or decrease in cross sectional area of the column.

Preferably in order to facilitate the outward movement of the burnt gases and hinder their return towards the cylinder, the exhaust ducts will be increased progressively in cross section in the direction of exhaust.

This progressive increase in cross section will allow for the expansion of the burnt gases during their motion away from the cylinder.

A suitable arrangement for carrying out the above requirements is indicated in Figures 11 and 12. In these figures it will be seen that the passage 4 in the cylinder block connecting the exhaust orifices 3 with the inlet end of the exhaust pipe 5 is so formed that it comprises no surfaces capable of reflecting the issuing gases back into the cylinder and that from its point of connection with the cylinder the exhaust pipe increases progressively in cross section.

In arranging the form and shape of the exhaust passages, account should be taken of the fact that the exhaust mass tends to be projected from the cylinder in a natural direction which is that of the cylinder axis, and it is for this reason that in Figure 12 the exhaust passage is inclined to the axis of the cylinder in order to deflect the issuing mass of burnt gases as little as possible.

A further object of the invention is to specify the relationship that should exist between the length of the exhaust pipe and the return impact of the burnt gases for the most advantageous operation of the engine.

It has been stated above that the inventor has found by experiment that when the burnt gases issue from the cylinder directly into the atmosphere, the return impact of the burnt gases occurs most rapidly and that the rapidity of the return impact is reduced when the burnt gases pass through an exhaust pipe before reaching the atmosphere.

He has observed that the delay and the intensity with which the return impact occurs is influenced within limits by a variation in length of the exhaust pipe.

Figure 6 illustrates the influence of a variation in length of the exhaust pipe upon the return impact. In this figure three curves are shown which are all taken at the same engine speed, but with exhaust pipes of different lengths fitted to the engine, the length of the exhaust pipes being 2 feet 6 inches (chain dotted curve), 4 feet 5 inches (full line curve), and 5 feet 8 inches (dotted curve), respectively.

The curves shown in Figure 6 are similar to those represented in Figures 1 to 5, but indicate pressure variations only.

It will be seen that as the length of the pipe increases, the return impact becomes more distant, a longer period of time is available for

charging and consequently the effect of the return impact upon the cylinder is reduced in intensity.

If the length of the exhaust pipe continues to be increased, a point is reached when no further retardation of the return impact can be obtained.

If the exhaust pipe is of tapered form flaring suitably outwardly, then a further increase beyond this point will not be objectionable, since the continued increase in cross section of the exhaust pipe will permit the expansion of the gases to occur, but such an increase in length will be unnecessary. If, on the other hand, the exhaust pipe is cylindrical in form, then by lengthening the exhaust pipe still further after the point of maximum retardation has been reached, the return impact will again commence to occur sooner and this is due to the fact that the cylindrical form of the pipe will not permit the free expansion of the gases and produces a choking effect which opposes the final exit of the gases after their expansion.

If a silencer or expansion chamber is fitted on the exhaust pipe, then in the case when the exhaust pipe is of tapered form, such a silencer may be fitted at the end of an exhaust pipe of substantially the required length to give the maximum retardation in the return impact, but as will be seen from the above remarks, the silencer may be fitted at the end of a suitably tapered exhaust pipe of more than the necessary length without objection.

In the case when the exhaust pipe is cylindrical in form, the silencer should be fitted at the end of an exhaust pipe of substantially the required length to give the maximum retardation, and if any further lengthening of the exhaust pipe is required, this should be provided beyond the silencer or expansion chamber.

By way of indication it may be stated that in general it will be found that a length of exhaust pipe from 3 feet to 6 feet will be found sufficient for the above purposes.

In explanation of the above it may be stated that the volume of the void left in the cylinder and the exhaust pipe by the mass exit of the burnt gases will depend upon the mass of the external gaseous medium that can be displaced by the work the exhaust gases are allowed to do by impact upon this external medium, other things being equal.

The time absolute for which this void lasts is dependent upon the length of the path travelled by the exhaust gases before they rebound towards the cylinder, or more generally upon the velocity of exit of the burnt gases and the negative acceleration they then undergo.

If the exhaust gases are allowed to issue directly from the exhaust orifice into the open atmosphere, the head of the issuing column of gases will be flattened and enlarged. The resisting surface will thereby be increased in area and the negative acceleration applied to the issuing mass will be extremely high. Consequently the path travelled by the gases from the cylinder will be extremely short and the rebound into the cylinder will follow with extreme rapidity.

If on the other hand the gases have to pass through an exhaust pipe or the like before reaching the open atmosphere, and this pipe is too short to contain the column of exhaust gases, the head of the latter is still crushed and deformed against the atmosphere external to the pipe so that the time elapsing between the exit and the return of the exhaust gases is reduced in relation to the extent of this deformation of the column.

If the exhaust pipe is lengthened, a dimension will be reached for which for a given explosive force the rebound of the burnt gases will occur from a plane or frontal zone situated adjacent the end of the pipe. After this length of the pipe has been reached, the further lengthening of the pipe will not produce any additional delay in the occurrence of the return impact. In other words, there will be no advantage in lengthening the pipe beyond this point.

If silencers or expansion chambers are provided on the exhaust duct, their position will be determined exactly by the point from which the rebound of the burnt gases occurs for the strongest explosions.

At explosions of high intensity the energy contained in the burnt gases and their exit velocity are greater and consequently the point from which they rebound is situated more distant from the engine than for explosions of lower intensity.

For weak explosions, therefore, everything will occur as if the exhaust pipe were too long, in other words, without advantage from the point of view of the present invention.

Expansion chambers will however become objectionable if they are situated too near the cylinder, that is to say, nearer the cylinder than the furthest point of rebound of the column of exhaust gas for the strongest explosion over the working range of the engine.

Such a sudden increase in section of the exhaust pipe as a consequence of the provision of an expansion chamber or the like, nearer than the point of rebound of the burnt gases, would have a similar objectionable effect to that which occurs if the point of rebound is allowed to become situated in the open atmosphere. In other words, the point of rebound would become formed in the expansion chamber itself, the return would consequently occur in a reduced space of time and the period available for charging would be shortened.

According to the invention therefore, the characteristics of the exhaust system are so related to the energy contained in the burnt gases upon their exit from the cylinder at maximum intensity of explosion that the return impact of the said burnt gases always occurs from a point substantially within the said exhaust system, the shape and form of the said exhaust system being such that it comprises no sudden changes in section and that the issuing mass of burnt gases is always thereby guided and directed away from the cylinder.

This result may be attained by providing the engine with an exhaust pipe of such a length that the return of the burnt gases for an explosion of maximum intensity occurs from a point situated substantially within the said exhaust pipe, and the requisite length of such an exhaust pipe can readily be determined by trial, as will be understood from the foregoing.

Further, when the engine comprises a silencer or expansion chamber according to the invention, such silencer or expansion chamber will be situated on the exhaust system or duct at a point more remote from the cylinder than the point from which the return impact of the burnt gases occurs.

The above considerations concerning the form and length of the exhaust passages and ducts apply equally well to a single cylinder and multicylinder engine, but in determining the form and

arrangement of the exhaust ducts and manifolds for a multi-cylinder engine further considerations arise as will be understood from the following.

In an internal combustion engine according to the invention, normally the exhaust, the passage of fresh gases through the cylinder and the end of the charging will occupy about 120° of the crank movement.

According to the relative timing of the cylinders, the total phases of exhaust and charging may overlap.

It may occur that the exit from one of the cylinders is produced at the same time as an admission occurs into another cylinder and according to the number of cylinders the phases of admission and exhaust may themselves overlap.

Normally for three cylinders grouped together on a crank shaft, these total phases of outlet of the burnt gases and entry of the fresh charge are separated in time or overlap very little according to the timing of the engine.

With the timing represented in Figures 1 to 5, for example, in a three cylinder engine there will be an overlap of a few degrees.

In engines with four or more cylinders all these phases overlap.

When establishing the exhaust manifolds for a multi-cylinder engine, care must be taken to establish a protection against disturbances that may be produced by the exhaust from one of the cylinders upon another cylinder.

The burnt gases upon their mass exit from the cylinder and also upon their rebound towards the cylinder, behave like a projectile and are governed by the laws of reflection.

Consequently if the burnt gases upon their outward passage from one cylinder encounter surfaces which tend to reflect them towards another cylinder in which the exhaust and inlet orifices are open at this moment, this may have an objectionable effect upon the charging of the latter.

Consequently in arranging the junctions between the ducts and manifolds in a multi-cylinder engine, care will be taken to ensure that the walls of the ducts and junctions are profiled in such a way that the angles of reflection always lead the gases to the exterior and away from another cylinder which may be open at this moment.

Further, in establishing the exhaust manifolds and exhaust pipes, the same rules must be observed as for an exhaust pipe of a single-cylinder engine, that is to say, there must be no sudden and considerable increase or decrease in cross section on the path followed by the gases as this will cause the exit speed to be reduced and a total or partial rebound may be produced which may influence the cylinder which is exhausting or one of the cylinders in which the exhaust and inlet orifices are open.

In addition when the mass of burnt gases leaves the cylinder during the exhaust period, and when it is passing a junction which is also connected with a cylinder open to admission at this moment, and in which exhaust is still open, the mass of burnt gases in passing the junction causes a shock to be transmitted along the exhaust duct leading from the second mentioned cylinder and this shock may have an objectionable repercussion on the admission of the charge into the latter cylinder.

This shock will be transmitted for a certain distance, after which its effect will be no longer noticeable. This objectionable effect may be overcome by making the distance between the junction and the second cylinder of a length at least equal to the minimum distance that will ensure that the objectionable effect mentioned above will not be transmitted to this cylinder.

A suitable distance for any particular case will be found by trial, but as a general guide it may be mentioned that a distance of 20 to 30 cms. will generally be sufficient for an engine of 1 litre capacity per cylinder.

Alternatively, reflecting baffles of the form described and shown in application Serial No. 738,016 filed August 1, 1934 may be provided at the junctions between the ducts leading from two cylinders, in order that the effect of any such shock will be minimized.

Care should also be taken to combat any objectionable action which may be produced by a prolonged suction caused by the exit of the burnt gases from one cylinder upon the charging of another cylinder. This may be avoided by a suitable closure of exhaust, as described in application Serial No. 84,184 filed June 8, 1936.

Further, the return of the burnt gases which have left one cylinder may have an objectionable effect upon the charging of another cylinder and a protection should be afforded against such objectionable action.

This protection may also consist in establishing a suitable closure of exhaust as described in application Serial No. 83,120 filed June 2, 1936 for the cylinder to be protected or by the provision of the reflecting baffles of application Serial No. 738,016 referred to above which redirect any returning gases in the direction of exhaust or by any other suitable means which ensure that the said objectionable action will be combated.

In establishing the junctions between the exhaust ducts and manifold of two cylinders or of two groups of cylinders, account must also be taken of the state of the gases at the junctions and the ducts which will be traversed by the exhaust gases. Two extreme conditions may be produced. Either the exhaust gases issuing from one cylinder may encounter the return impact of the preceding exhaust or the exhaust gases may enter a void left by the preceding exhaust.

If it is desired to reduce the absolute time of exhaust, an arrangement may be adopted such that each exhaust from one cylinder falls into the void left by exhaust from the other cylinder. The same considerations apply to the columns of exhaust gases, and the columns of entering gases which follow the exhaust columns. Those columns which have the same direction of movements do not exert any resistance upon each other, on condition that the spaces and the sections of passage permit them to intermingle without too much deformation, and it is logical to consider that the angles at which such columns encounter each other must be reduced to a minimum.

It should be noted that with a multi-cylinder engine the following effects may be obtained by arranging the exhaust ducts in accordance with the foregoing:—

1. That every exhaust will occur through a duct containing gases which are under a depression and which are moving in the same direction as the exhaust. This current of gases under depression is formed by the admission gases which pass through the cylinder of one of

the cylinders adjacent the cylinder which is under exhaust.

2. That exhaust gases moving outwards through the piping may encounter returning gases from the preceding exhaust and may arrest these returning gases at a constant distance from the cylinders.

By means of the invention, the cylinders may be protected against any disturbances from the adjacent cylinders by the length of the first junction, by the surfaces which reflect and guide the gaseous column, by the sections permitting the free passage of the columns without restriction and without crushing against the ambient mass, by the position of the junction of the manifolds. This position is determined by the position of the returning column of gases from the preceding exhaust, the length, the volume of the exhaust duct after these first junctions and the position and volume of the silencer or expansion chamber.

The above remarks will be more clearly understood by referring to Figure 13 which is a timing diagram for a six-cylinder engine, each cylinder of which has the inlet and exhaust events established as indicated in Figures 1 to 5. In this figure it is assumed that the firing order is 1—5—3—6—2—4 by way of example. The figure shows simply the 60° intervals between the exhaust openings and the order in which such openings occur.

It will be seen from this figure and by referring to Figures 1 to 5 that the exhaust from one cylinder always occurs in the middle of the admission period of the preceding cylinder and may have a repercussion on this admission.

By making a manifold for each group of three cylinders 1, 2, 3 and 4, 5, 6, and extending these manifolds a sufficient distance before connecting them together as indicated in Figures 14 and 15, it will be easy to ensure that the above objection will be avoided.

But by referring to Figures 1 to 5 it will also be seen that at all speeds above 800 R. P. M., the return of the burnt gases following the exhaust from one cylinder, occurs after the opening of the exhaust of the succeeding cylinder.

If the two manifolds are extended to form separate exhaust pipes or if the junction between them is more remote from the cylinder than the point from which the return takes place, then no advantage can be obtained from the fact that the exhaust from a cylinder in one manifold occurs before the return towards the preceding cylinder in the other manifold.

On the other hand, if the manifolds are connected to a common exhaust pipe as shown in Figure 15 and the junction between these manifolds is situated nearer the cylinders than the point in the piping from which the nearest return of the burnt gases occurs then the exhaust gases from one cylinder, say cylinder 1, will pass this junction and will not have commenced to return at the moment when the exhaust of the next cylinder, say cylinder 5, commences so that the exhaust from the latter cylinder will enter piping which is under depression as stated above.

These exhaust gases from the cylinder 5 will pass the junction and will oppose the returning exhaust gases from cylinder 1, so that when the next cylinder, say cylinder 3, opens the exhaust gases from the latter will also enter a depression in the piping and so on.

At high speeds the effect of this interaction between the exhaust gases may be such as to annul the effect of the return of the exhaust gases completely.

Moreover, and as explained in connection with the exhaust pipe of a single cylinder engine, the length of exhaust pipe following the junction between the manifolds should be sufficient to ensure that the return of the burnt gases occurs from a point situated within this pipe, whereby the return of the burnt gases will be delayed as much as possible and if a silencer or expansion chamber is fitted it will be fitted on this pipe, not nearer the cylinder than the point from which the furthest return occurs for a maximum intensity of explosion.

In practice the cylinders of a six or eight cylinder engine may be divided into two groups, the cylinders of each group exhausting into a common manifold. These two manifolds may extend over a distance approximately equal to the length of the engine itself and thereafter connected together so as to lead into a single exhaust pipe upon which is mounted a silencer device common to all the cylinders.

A suitable construction and arrangement of the exhaust piping of a six cylinder engine having the above timing and constructed in accordance with the above modifications is illustrated in Figures 14 and 15.

Cylinders 1, 2 and 3 are connected to one manifold 6 and cylinders 4, 5 and 6 to the second manifold 7 and these manifolds are connected at 8 to a simple exhaust pipe 9 upon which is mounted a silencer 10.

It will be seen that the main walls of all the passages are so formed that they tend to guide and direct the exhaust gases towards the exterior and so that they do not obstruct or restrict the outward passage of the gases.

It will also be understood that in this example, if the junction 8 is sufficiently remote from the cylinders to protect the charging of cylinder 3 from a repercussion of the exhaust from cylinder 6, this will ensure that all the other cylinders are sufficiently protected in the same manner. It will also be understood that the junction 8 must be nearer the cylinders than the point from which the return of the burnt gases occurs, and that the silencer 10 must be more remote from the cylinders than this point.

Having now described my invention, what I claim as new and desire to secure by Letters Patent is:

1. Method of controlling variable speed two stroke cycle internal combustion engines, which comprises establishing communication between the cylinder and exhaust system during the firing stroke, providing at all engine speeds over a chosen speed range, for the issuance of the burnt gases from the cylinder substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gas moves outward and thereafter returns from a point which may be within the exhaust system, providing a permanent free passage for the burnt gases to the limit of outward travel of said burnt gases, preventing at the highest engine speed the entrance of fresh charging air until the said issuance of the burnt gases is in full progress, admitting fresh charging air into the cylinder, at the highest engine speed, when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, thereby ensuring that at all lower engine speeds the said admission will not

commence until the said issuance of the burnt gases has occurred, and providing at all engine speeds for the said fresh charge to occupy the cylinder and a portion of the exhaust system in the interval elapsing between the said exit of the said gases and the instant when the pressure of the returning gases becomes effective within the cylinder.

2. Method of controlling variable speed multi-cylinder two stroke cycle internal combustion engines, which comprises establishing communication between each cylinder and its exhaust system during the firing stroke in said cylinder, providing at all engine speeds over a chosen speed range for the issuance of the burnt gases from each of said cylinders substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gas moves outward and thereafter returns from a point which may be within the exhaust system, providing a permanent free passage for the burnt gases issuing from each cylinder to the limit of outward travel of said gases, providing a communication between a plurality of the said free passages, and providing for the said burnt gases issuing from one cylinder to encounter in its free passage a depression caused in this passage by the issuance of burnt gases from another cylinder, preventing at the highest engine speed the entrance of fresh charging air into each cylinder until the said issuance of the burnt gases is in full progress, admitting fresh charging air into each cylinder at the highest engine speed when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, thereby ensuring that at all lower engine speeds the said admission will not commence until the said issuance of the burnt gases has occurred, and providing at all engine speeds for the said fresh charge to occupy the cylinders and a portion of the exhaust systems of said cylinders in the interval elapsing between said exit of the burnt gases and the instant when the pressure of the returning gases becomes effective within the cylinders.

3. A variable speed multi-cylinder two stroke cycle internal combustion engine having exhaust and inlet orifices on the cylinders and an exhaust conduit on the said orifice of each cylinder, the said exhaust conduits being connected together in groups to a common manifold, means for so controlling the exhaust orifices of the several cylinders as to ensure the issuance of the burnt gases substantially as a mass at all engine speeds above a chosen maximum speed, whereby the said mass moves outward and thereafter returns toward the cylinder, means for so controlling the several inlet orifices as to ensure, at the highest engine speed, that the said inlet orifices will be opened while the corresponding exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the said conduits and manifolds providing a permanent free passage for the burnt gases from the several cylinders to the limit of outward travel of said gases, the several conduits being connected together in a group having a sufficient length and being so formed internally at their connection to the manifold as to provide a passage which guides the burnt gases away from any other cylinder of the group which is open at the same time, and to protect the said other cylinder from an objectionable action by said issuing gases.

4. An engine as claimed in claim 3, a plurality of manifolds being connected together to deliver into a single exhaust pipe, the said connection being established at a point situated nearer the several cylinders than the shortest limit of outward travel of the burnt gases from any of said cylinders.

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Filed June 8, 1936 8 Sheets-Sheet 1

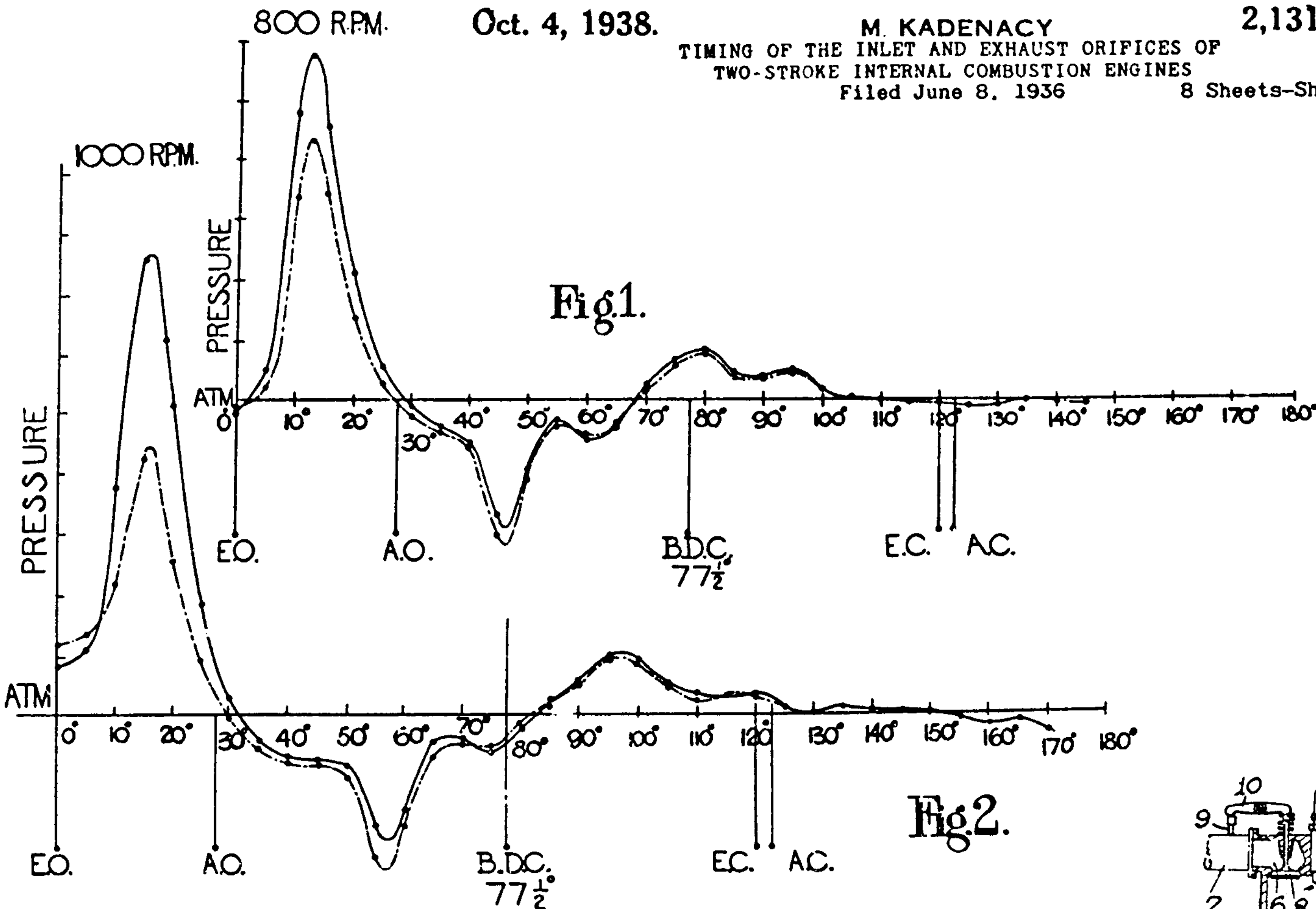

Fig.1.

Fig.2.

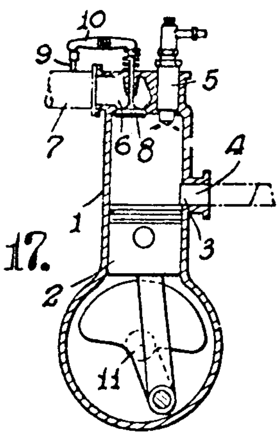

Fig.17.

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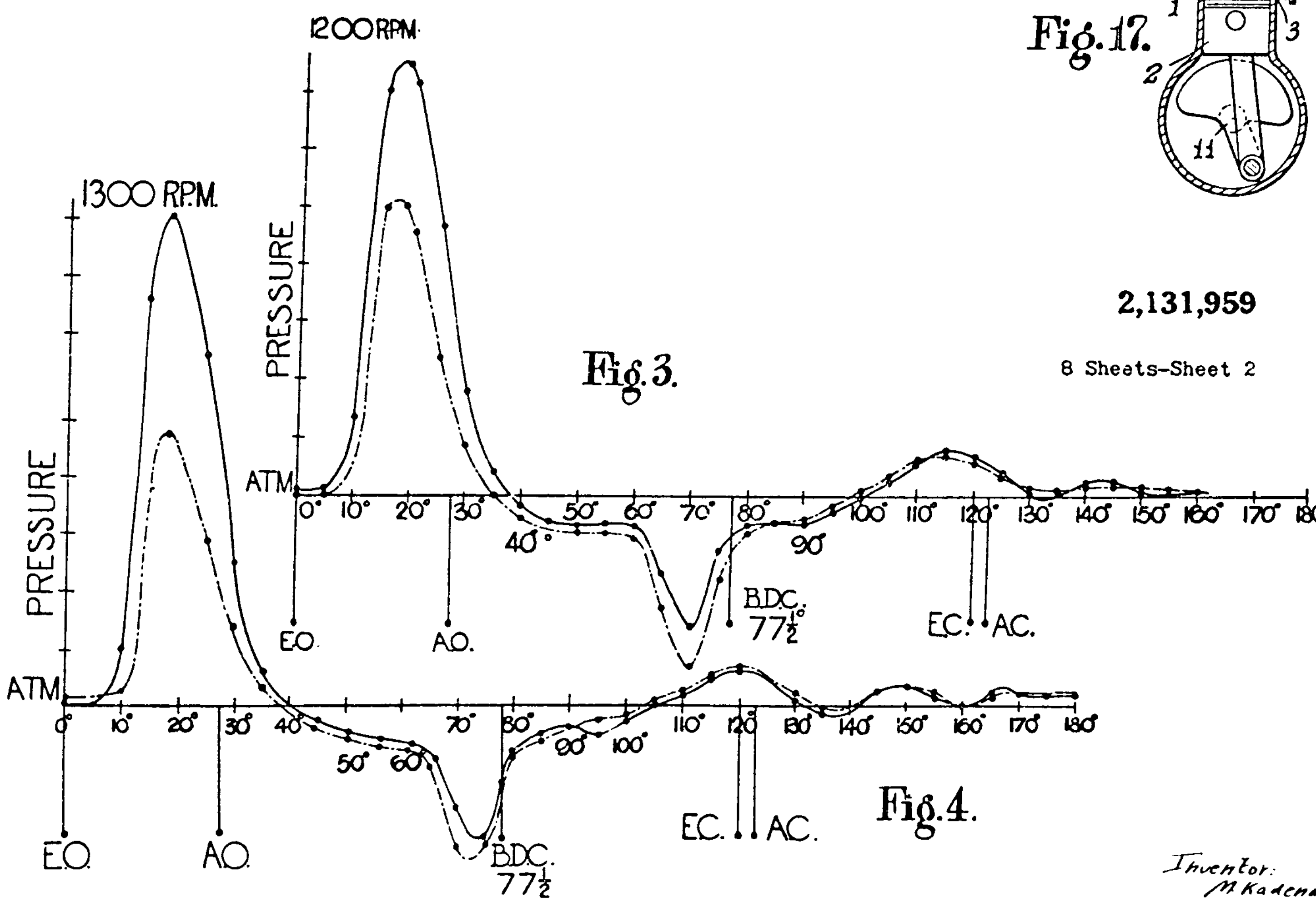

Fig.3.

Fig.4.

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TIMING OF THE INLET AND EXHAUST ORIFICES OF TWO-STROKE INTERNAL COMBUSTION ENGINES

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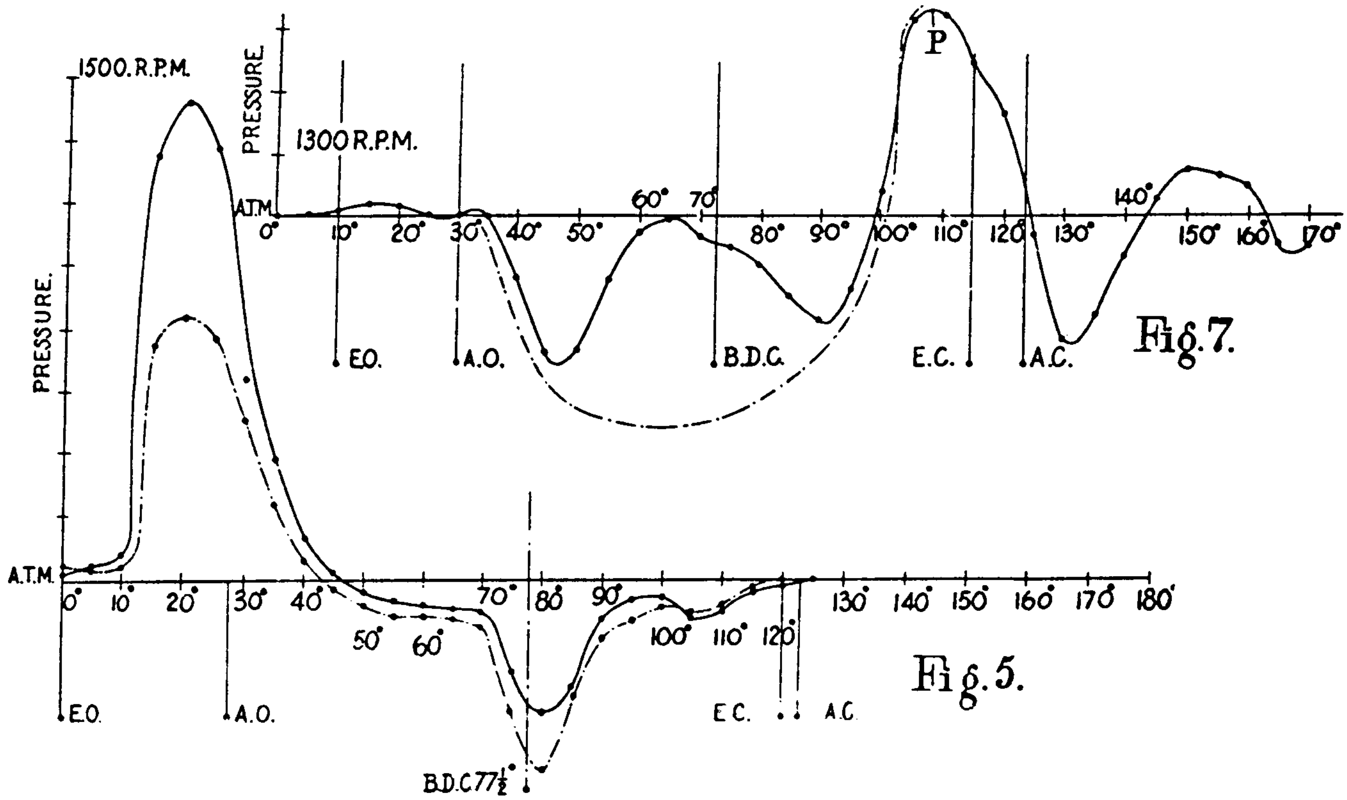

Fig.7.

Fig.5.

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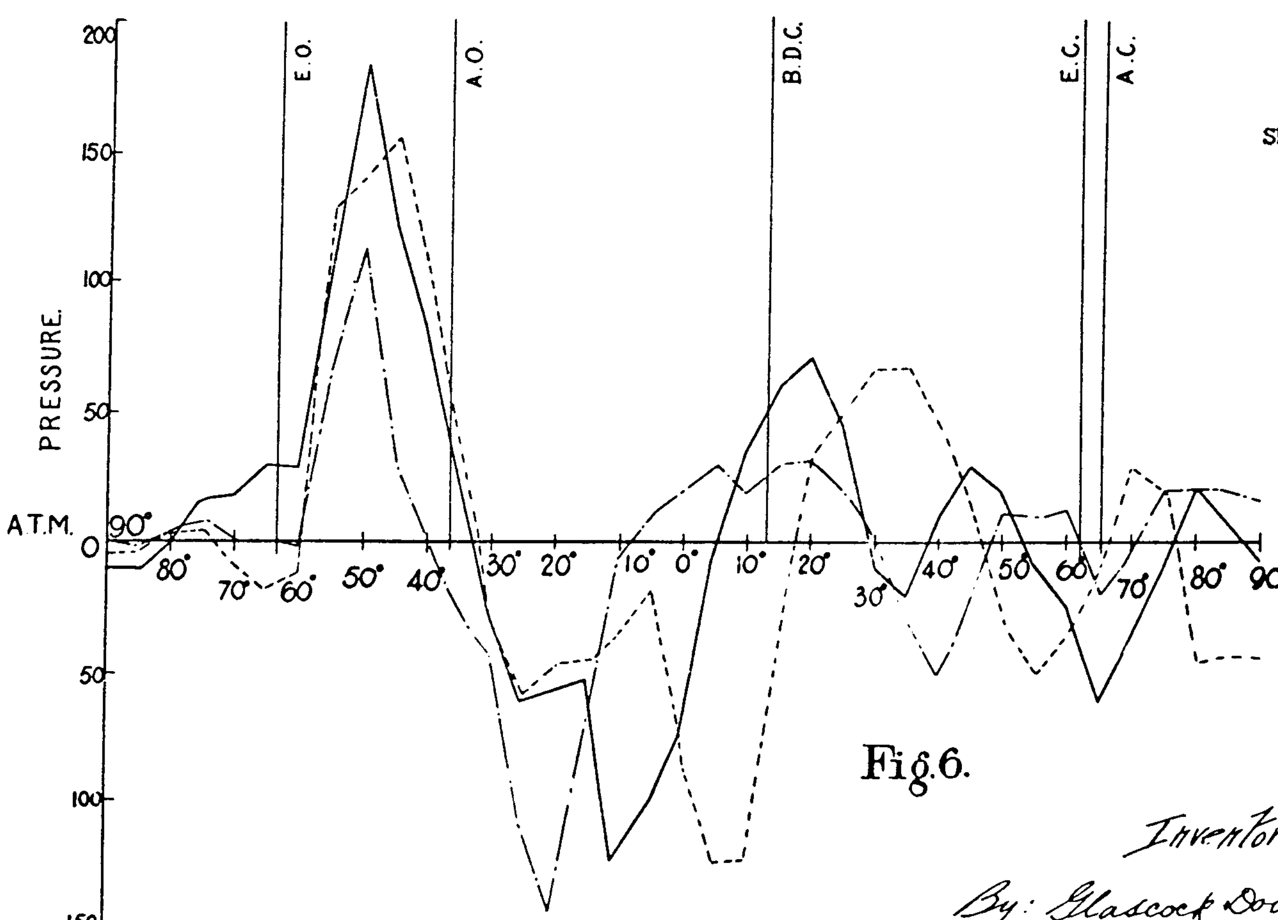

Fig.6.

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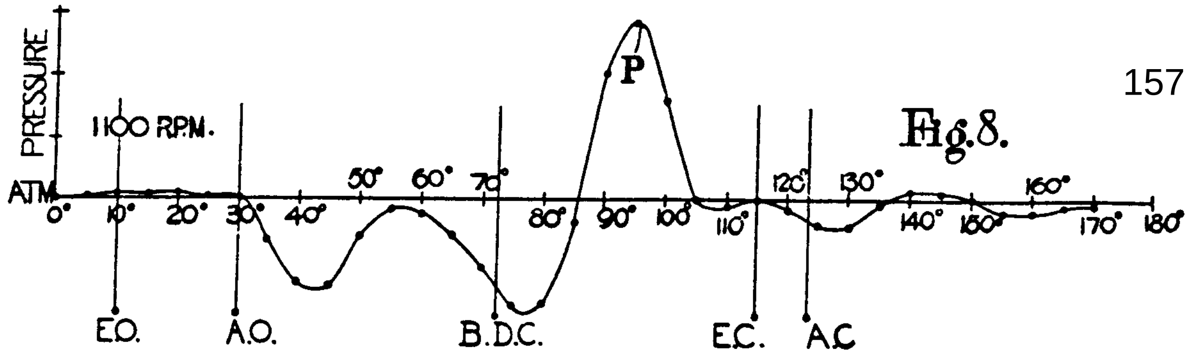

Fig. 8.

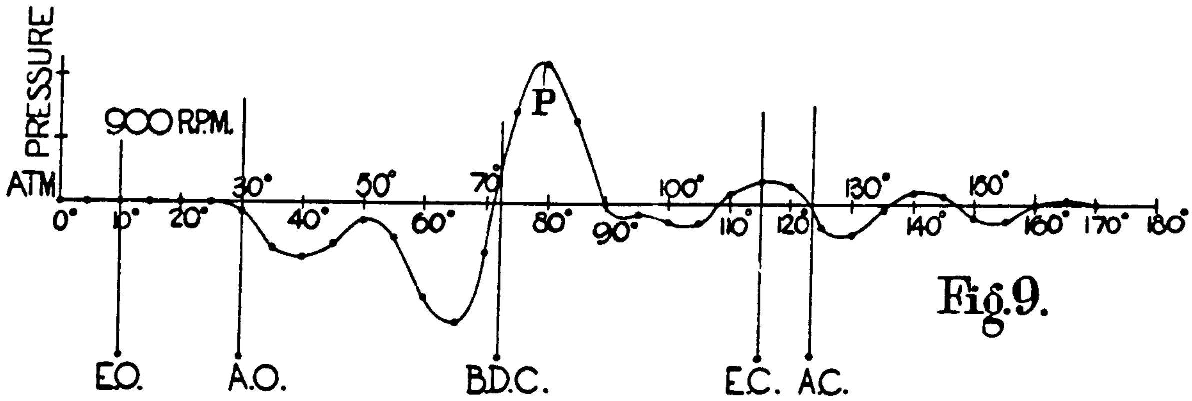

Fig. 9.

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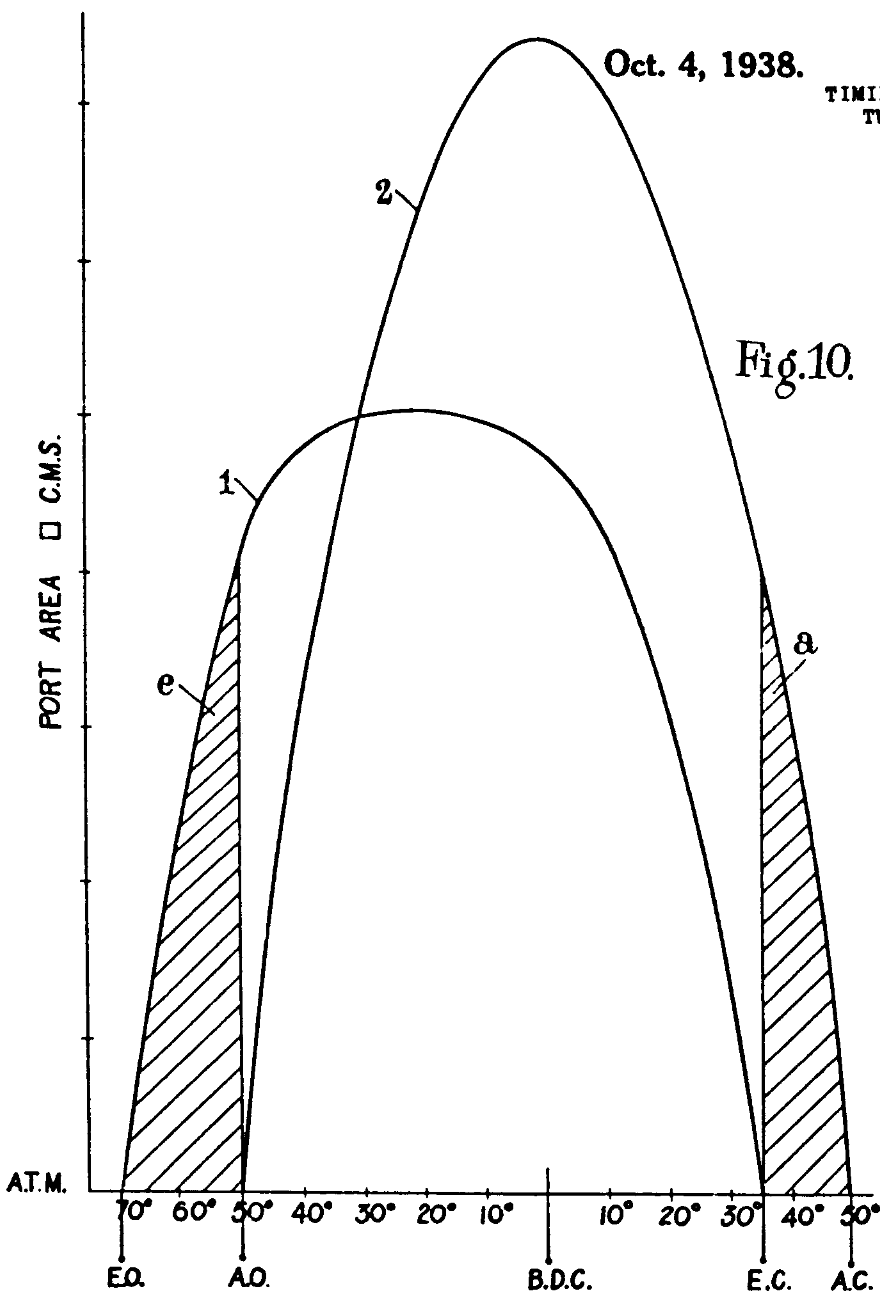

Fig. 10.

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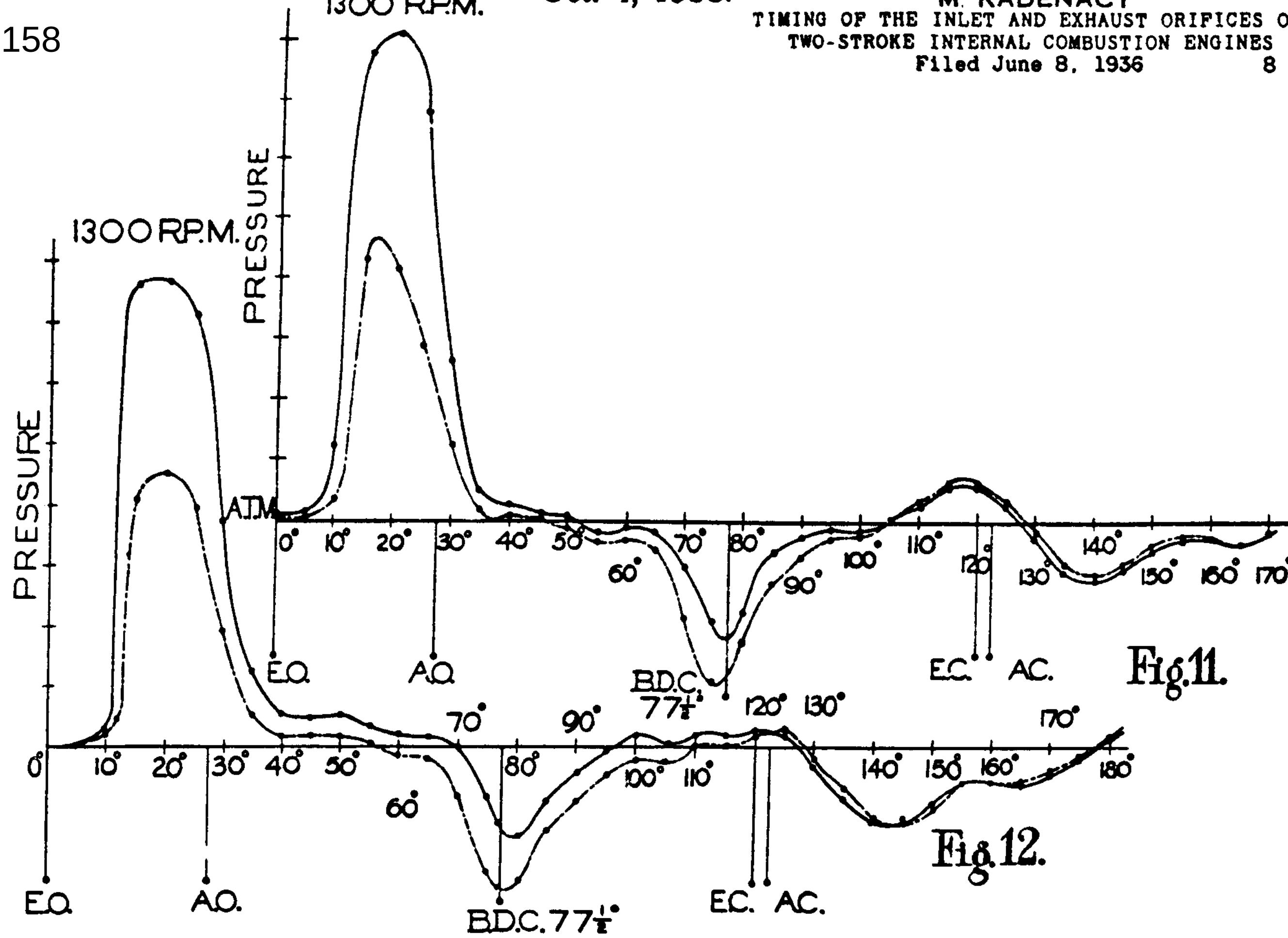

Fig. 11.

Fig. 12.

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Fig. 16.

Fig. 15.

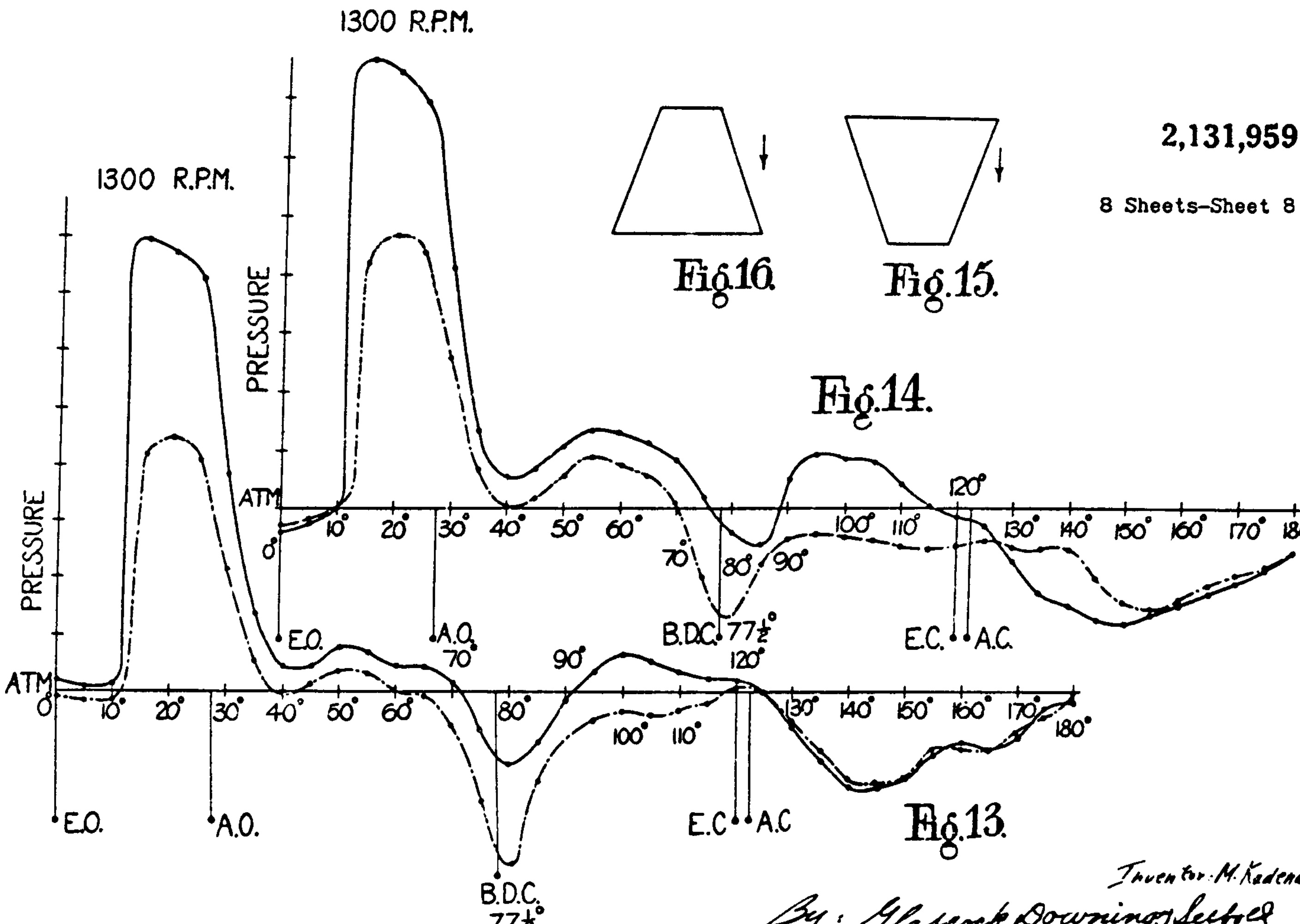

Fig. 14.

Fig. 13.

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Patented Oct. 4, 1938 2,131,959

UNITED STATES PATENT OFFICE

2,131,959

TIMING OF THE INLET AND EXHAUST ORIFICES OF TWO-STROKE INTERNAL COMBUSTION ENGINES

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Application June 8, 1936, Serial No. 84,184
In Great Britain January 11, 1936

3 Claims. (Cl. 123—65)

This invention relates to a method of constructing two-stroke cycle internal combustion engines of the kind wherein at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when an adiabatic flow only is involved, and in such a short interval of time that it is discharged as a mass leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder by opening the inlet orifice with the required delay after the opening of the exhaust orifice to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct and that a suction effect is exerted at the inlet orifice as a consequence of the exit of the said mass.

In application Serial No. 84,182 filed June 8, 1936, the characteristics that should be possessed by the exhaust and inlet orifices of such engines in order to obtain a maximum output at varying speeds have been specified. It is indicated in the said specification that the exhaust and inlet orifices must be made as large as possible within mechanical limits and that the full area of the said orifices must be opened as rapidly as possible and that the inlet orifice should open as close as possible to the moment of exit of the burnt gases to provide a permissible opening of inlet over a speed range of the engine, whereby the maximum charge may be introduced into the cylinder in the time available; and it is also indicated in the said specification that provision should be made to ensure that the charge will be retained in the cylinder and will not be forced out or sucked out by movements of the exhaust gases.

In one embodiment of the invention, the inlet is arranged to open at a constant crank angle after the commencement of exhaust opening, this crank angle being so chosen that at the varying engine speeds, inlet will always open at a suitable moment for the introduction of the charge and as close as possible to the moment of exit of the burnt gases, the exhaust system being such that the outward movement of the burnt gases will continue for a sufficient length of time to permit the said timing to be established, and means being provided in the exhaust system which cause the duration of time of exhaust to be reduced progressively as the speed of the engine increases, whereby the increase in the angle occupied by the exhaust at increasing engine speeds can be reduced to a minimum.

In a further embodiment of the invention the exhaust is arranged to close after a constant crank angle from the commencement of exhaust or from a suitably chosen point on the rotation of the crank shaft, this angle being so established that exhaust will always close, at the varying engine speeds, when the inlet has opened and the incoming charge is passing through the cylinder, after the exit of the burnt gases from the cylinder has been completed and before any return impact occurs, a suitable exhaust system being provided which ensures that the outward movement of the burnt gases will continue for a sufficient length of time to permit the said timing to be established.

As in the case of inlet opening at a constant crank angle, this closure of exhaust at a constant crank angle will also be an arbitrary choice.

In a still further embodiment of the invention, in order to enable the time of passage of the incoming charge through the cylinder to be constant in its duration at the varying engine speeds, the exhaust is arranged to close at a constant time interval after the commencement of exhaust opening, whereby the exhaust will always close at the varying engine speeds at the same moment in time absolute relative to the return of the burnt gases, a suitable exhaust system being provided to ensure that the outward movement of the burnt gases will continue for a sufficient length of time to permit the said timing to be established.

The invention will now be described in greater detail with reference to the accompanying drawings.

In the drawings:

Figures 1 to 5 are curves of pressures taken on the exhaust duct of an internal combustion engine at varying engine speeds.

Figure 6 shows three curves of variations of pressure in the exhaust duct for the same engine speed but with different lengths of exhaust pipe.

Figures 7, 8 and 9 show three curves of pressures taken on the inlet duct of an engine, and

Figure 10 is an exhaust and inlet port area diagram.

Figures 11 to 14 are curves of pressures taken on an engine at the same speed but with the use of a source of compressed air for charging the engine.

Figures 15 and 16 are diagrammatic representations of exhaust port openings serving to illustrate the invention.

Figure 17 shows, by way of example, a two-stroke cycle internal combustion engine to which the present invention has been applied.

In all of Figures 1 to 14, EO, EC, and AO, AC represent exhaust opening and closing and inlet opening and closing respectively, and BDC indicates bottom dead centre.

The curves shown in Figures 1 to 9 and 11 to 14 were obtained by means of the pressure indicating device described in application Serial No. 82,958 filed June 1, 1936, and the curves shown in Figures 1 to 5 and 11 to 14 are double pressure records taken with an indicator device comprising a double Pitot tube in operative communication with a manometer, the pressure read-

ings being taken in synchronization with the engine cycle and being adjustable relative to the cycle, thereby enabling direction of movement and the velocity of the gases to be observed or determined.

It will be noted that each of Figures 1 to 5 and 11 to 14 comprises two curves, one in full lines and one in chain dotted lines. Where the full line curve is above the dotted line curve, it is to be understood that the movement is in the direction leading away from the cylinder. The pressure difference between the ordinates of the two curves at any particular crank angle is also a measure of the velocity with which the gases are moving at this moment.

From a consideration of the double pressures shown in any of Figures 1 to 5, it will be seen that after the commencement of exhaust opening, a delay occurs during which substantially no pressure difference exists and during which the direction of movement of the gases may be towards or away from the cylinder.

Thereafter the pressure difference increases rapidly to a maximum value which remains substantially constant for a time period which is substantially the duration of passage of the column of exhaust gases past the indicating device.

The movement then continues in an outward direction and the cylinder and exhaust piping remain under a depression while the entering admission current is passing through.

The curves also indicate in a very clear manner the moment when the movement of the exhaust gases and the admission gases which follow them is suddenly reversed in direction.

The delay that occurs between the commencement of opening of the exhaust and the moment when the column of burnt gases engages in the exhaust duct may be explained as follows:

When the exhaust orifices commence to open the total area opened is too small for the gaseous body formed by the exhaust gases to engage therein.

After a certain lapse of time this area becomes sufficient for the column to engage therein with its ballistic velocity and this is the true moment of commencement of the mass exit of the burnt gases from the cylinder. The delay in the commencement of the mass exit of the burnt gases is further due to the inertia of the gases which are moving in the cylinder at the moment of exit and this inertia must be understood as a change of direction of movement of the gases and not a change from a state of rest to a velocity.

The curves show that when the burnt gases engage in the exhaust duct the rise in pressure to a peak and the consequent fall are extremely abrupt.

The speed of exhaust, that is of the mass exit of the burnt gases is determined by the energy of the gases at the moment of exhaust. The curves show that this speed is very high and everything occurs as if the gases already had an initial velocity in the cylinder.

The duration of exhaust, that is to say, the duration of the passage of the mass of burnt gases through the exhaust orifice depends on further factors, if the speed of the burnt gases is considered to be constant for the moment, since this initial speed varies according to the intensity of the explosion per unit mass.

The first factor is the rate of opening of the orifice through which the exhaust is effected. It is advantageous for this rate to be as great as possible since in this way the length of the column formed by the issuing mass of burnt gases will be shorter and the duration of its passage through the exhaust orifice will be reduced.

Secondly, the medium which the mass of burnt gases encounter on their path from the cylinder exerts a resistance on account of its inertia. The smaller the volume of this medium, within limits, the smaller will be the negative acceleration of the mass of burnt gases, in other words they will retain their velocity longer when this resistance is small.

Further, the mass of burnt gases possesses viscosity and if the exhaust orifice has the shape of a slit, this will produce too great a deformation of the gases which will lag in penetrating the exhaust duct and in becoming transformed into a column.

Similarly, if the exhaust duct beyond the exhaust orifice has too small a cross section relative to the area of exhaust orifice open at the moment of exit of the burnt gases, this will exert a further compression and braking action on the burnt gases.

On the other hand, if the exhaust duct following the exhaust orifice is too large relative to the area of this orifice, the head of the column is crushed or flattened and the impact surface of the column increases in size, thereby reducing the path travelled before the rebound occurs.

The curves shown in Figures 1 to 5 were all taken on the same engine with the same exhaust system. These curves enable us to see the absolute duration of time during which the exhaust occurs, the duration in absolute time for which the depression, or the current of entering air, lasts in the cylinder or exhaust piping and the moment of reversal of the direction of movement of the gases, all these times being measured from the exact moment at which the exhaust orifice is opened.

From these curves it can be seen that for a definite arrangement of the exhaust system the complete exit of the burnt gases from the cylinder occupies a substantially constant lapse of time from the moment when the exhaust orifice commences to open, but that this lapse of time tends to decrease with increasing speeds due to factors related to the rate of opening of these orifices or the modification in the resistance afforded by the gaseous medium present on the path of the mass of burnt gases.

The curves also show that the moment of reversal in direction of movement of the gases is separated from the commencement of exhaust or in practice from the commencement of opening of the exhaust orifices, by a substantially constant duration of time.

The duration of the vacuum or of the admission current is also substantially constant in time. This can also be seen from Figures 7, 8 and 9.

All this is independent of the speed of rotation of the engine, other things being equal, and the inventor has also found that variation in intensity of the explosion in the engine varies these times only in an insignificant manner.

In establishing an engine in accordance with the invention the above phases must be taken into consideration in order to ensure that the engine may be supplied with its charge during the angular sector which has been chosen for evacuating and charging the engine and outside

which the cylinder is closed completely for the compression and working strokes of the piston.

The filling of the cylinder is dependent upon the area of the charging orifices, the time of opening of these orifices and the differences in pressure which exist in the inlet ducts and the cylinder itself and also in the exhaust ducts which communicate directly with the cylinder when charging takes place.

It is useful to note that it is not only the static pressures that are important and that the velocity acquired by the entering charge also enters into consideration. During the admission period, the column of entering charge follows the column of exhaust gases which is moving in the direction of exhaust and which may itself be under a very high static pressure and irrespective of the static pressure within the exhaust column, the volume of the void left behind this column in the cylinder and in the exhaust ducts may increase at a higher rate than it can be filled by the entering charge. The greater the volume of the void left by the issuing gases and the more delayed the return of these gases, the higher will be the velocity acquired by the entering charge.

In all the curves shown in Figures 1 to 5, the fall in pressure consequent upon the mass exit from the cylinder is partially arrested by the entering current of admission gases consequent upon the opening of inlet and the influence of the return impact of the burnt gases is reduced by the provision in the exhaust duct of the means described in application Serial No. 46,805, filed October 25, 1935, which cause the return of the burnt gases to be by-passed into a chamber other than the engine cylinder.

But it can be shown that the return of the gases in general occurs at a very high speed and that it is clearly defined by a dense frontal zone which confirms that its nature is that of a body that rebounds from the obstacle it has encountered.

It can also be shown that if no inlet is opened after the mass exit of the burnt gases occurs, a volume of vacuum is left in the cylinder and in the exhaust ducts and the exhaust gases after their rebound encounter no obstacle upon their return path towards the cylinder.

Consequently the return is much more vigorous, it enters the cylinder at a much greater speed, and the duration of the vacuum in the cylinder is reduced.

In order to obtain a maximum utilization of the vacuum left by the mass exit of the burnt gases it is therefore necessary for the admission to open immediately after the depression is formed adjacent the inlet and with the greatest rate area of opening per unit movement of the piston, as explained in application Serial No. 84,162 filed June 5, 1936.

If the admission orifices are opened at an opportune moment and immediately when the exhaust gases have left the point in the cylinder at which these orifices are situated and if the area and time of opening are appropriate a column of admission gases will enter the cylinder and follow the exhaust gases into the exhaust duct.

The more massive this admission becomes, the more effective will be its action in destroying or reducing the intensity of the return impact thus rendering it more easy to establish a closure of exhaust before the return of the burnt gases.

If the opening of the inlet orifices is delayed the return impact will return more quickly and with greater violence; the absolute time available for admission will be reduced and the disturbances produced by the return impact will be greater and more violent.

It should here be noted that this reasoning is strict from the theoretical point of view. In practice a fairly large tolerance is admissible from the constructional point of view for a perfect operation of the engine, but the above directions must be taken into consideration in designing the engine so that this direction will be approached as closely as possible.

The time interval occupied by the exhaust, that is, the time required for the evacuation of the cylinder varies in an engine according to the arrangement of the exhaust system.

It also varies according to the resistance of the medium in these exhaust systems. These factors have a very great influence on the operation of the engine. A great variation in speed of rotation of the engine also influences the area time factor so that the duration of exhaust varies. But in any case the duration of time of exhaust in an engine for a given exhaust system remains practically constant.

The inventor has found that by modifying the exhaust system the time of duration of exhaust can be varied. For example this result may be obtained by an adaptation of the area of the exhaust orifices and of the time during which they open progressively. This may be described by the expression area time factor. Another factor is the variation of rate of opening of the exhaust orifice.

The useful area time factor must be considered to be that which has served effectively from the commencement of the mass exit of the burnt gases until the evacuation of the cylinder, which period of evacuation is relative to the energy of this exhaust and the result must be ensured that at each engine speed the useful area time factor is that required for the purpose specified.

The duration of the vacuum in the cylinder and the exhaust piping is determined by the moment at which the return impact occurs at the chosen point in either component, for example at a point along the exhaust piping, or in the cylinder or at a point in the cylinder.

The inventor has established that the absolute time that separates the exhaust and the return impact may be very small, or may be more or less great according as the phenomenon of exhaust and of the return impact follow each other immediately and consecutively or are separated by an appreciable lapse of time.

Similarly, the inventor has found that the return impact may be destroyed in the exhaust piping (application Serial No. 38,826, filed August 31, 1935, application Serial No. 82,959 filed June 1, 1936); it may be reflected (application Serial No. 738,016, filed August 1, 1934) and may not penetrate into the cylinder proper, it may be opposed by an injection of air or gas into the exhaust system (application Serial No. 46,804, filed October 25, 1935) and so on. The inventor has established and confirmed by the said curves that the return of the exhaust gases for one and the same exhaust system occurs in the same absolute time. This time or rather the length of this time is expressed by the number of degrees proportional to the speed of rotation.

If the resistance offered by the gaseous medium contained in the exhaust piping is reduced, the speed of exhaust over its duration of time will be

greater and the time occupied for the complete evacuation of the cylinder will be smaller. In this case as is logical, the return will take longer to occur for the reason that the gaseous body will be projected for a longer distance, the position from which the return occurs will be further away, the initial speed of the gases being the same and the paths travelled being longer.

Further, by a suitable arrangement of the exhaust system, the reversal in direction of motion of the burnt gases can within limits be made to occur from a point more or less distant from the cylinder as desired.

For example Figure 6 shows the influence of altering the length of the exhaust pipe. This figure shows three curves taken on the same engine at the same speed and with three different lengths of exhaust pipe the length of the exhaust pipes being 2 feet 6 inches (chain dotted curve), 4 feet 5 inches (full line curve) and 5 feet 8 inches (dotted curve). It will be seen that as the length of the exhaust pipe increases the absolute duration of time of the vacuum or of the admission current is lengthened until a retardation which is a maximum for the exhaust pipe under consideration is reached, as is also explained in application Serial No. 84,182, filed June 8, 1936.

The series of curves taken at different speeds for the same arrangement of the exhaust system (Figures 1 to 5), the variation of these curves according to different arrangements of the exhaust system (Figure 6) and the variations in these curves as a consequence of the provision of specific arrangements (for example those described in applications Serial Nos. 38,826, 46,804, 46,805, 120,118, filed August 31, 1935; October 25, 1935; October 25, 1935 and January 11, 1937 respectively) introduced into these exhaust systems confirm the above statements.

A similar confirmation is obtained in the operation and output of the engine.

In establishing a fixed timing of inlet opening which will remain suitable over a chosen speed range of the engine and which is as close as possible to the moment at which the burnt gases leave the inlet orifice during their mass exit from the cylinder, the possibility of varying the above factors in order to obtain a desired result should be borne in mind.

For example by suitably varying the said factors it is easy to arrange so that the tendency of the time interval occupied by the total exit of the burnt gases to decrease with increasing engine speeds to be corrected so that this time interval is maintained more strictly constant. This may be effected by arranging or controlling the exhaust orifice in such a way that the area of exhaust orifice opened per unit movement of the piston or crankshaft decreases progressively as illustrated in Figure 15, and also by arranging the exhaust piping in such a way that the resistance to the mass exit of the burnt gases increases as the speed increases. For example the exit for the expanded gases from the silencer may be suitably restricted so that as the speed increases an increased resistance is created in the exhaust system.

Or the total exit of the burnt gases may be caused to be completed within a substantially constant crank angle, so that the inlet may be opened at the same position of the crank shaft for all working speeds of the engine.

For example the exhaust orifice may be so arranged or controlled that the area of exhaust orifice opened per unit movement of the piston or crankshaft increases progressively as illustrated in Figure 16, whereby the time period elapsing before the total exit of the burnt gases occurs may be reduced progressively as the engine speed increases.

A two-stroke cycle internal combustion engine embodying an exhaust orifice of this nature is illustrated in Figure 17, and comprises a cylinder 1 in which moves a piston 2 controlling an exhaust orifice 3 situated at the base of the cylinder and communicating with an exhaust duct 4. At the head of the cylinder is an injector 5 for the introduction of combustible fuel, and an inlet orifice 6 communicating with an inlet duct 7, and controlled by a poppet valve 8 by a push rod 9 and rocker arm 10 from the engine shaft 11.

A suitable arrangement of the exhaust piping will also assist in obtaining this reduction as the speed increases, for example by the provision in the exhaust pipe of the means described in application Serial No. 82,959 filed June 1, 1936, or, in a multicylinder engine, by a suitable arrangement of the exhaust manifolds as described in application Serial No. 84,182 filed June 8, 1936.

In the practical establishment of an engine it is more advantageous to arrange that the exhaust occurs over a constant crank angle than to arrange that exhaust occurs over a constant time interval since in the first case the admission orifices may be opened by the piston or by valve means controlled in fixed angular relation with the engine crank shaft, while in the second case the admission orifices must be opened by a distribution system employing a movement which is independent of the rotation of the engine, and extends over a constant lapse of time, but the commencement of the lapse of time will always start from the same point on the crank circle.

For example this may be done by means of a *slide valve with a stressed spring*. The release of this spring will be produced always for the same position of the piston for example. Consequently the initiation of the opening will be in direct relation with the position of the crank, but the opening itself will be produced in a constant lapse of time absolute which is independent of the speed of rotation of the engine. Instead of springs this may be effected by electro-magnets, by a hydraulic control or by a control utilizing gas under pressure, which is not important from the point of view of principle, but it is merely a constructional detail. The principle is that this opening is effected after a constant lapse of time after the commencement of opening of the exhaust as specified above.

The more the opening of the admission orifices is accelerated, the more advantageous it is for introducing the largest possible mass of fresh gases, and consequently for filling the void left by the exhaust gases and for opposing to the greatest possible extent the return of the burnt gases by the inertia of the entering charge, while a further advantage will be the retardation of the return impact due to the fact that the force exerted on the gaseous medium in the exhaust pipe by the burnt gases will be increased by the force exerted in the same direction by the entering charge.

From the double curves shown in Figures 1 to 5 it can be seen that after the end of the exhaust, the admission gases follow behind the exhaust gases and consequently they have passed through the cylinder, have occupied its entire volume, have swept through it and entered the

exhaust duct and continue to do so until the reversal in direction occurs which is produced by the return impact which sometimes arrives towards the end of the admission and sometimes even after the closure of inlet.

During these phases the cylinder although occupied with fresh air charge is under depression and it is advantageous to re-establish the pressure to the level of the atmosphere or to that which exists at the admission orifices outside the cylinder.

The volume left void in the cylinder and in the exhaust duct is very great, and it is not possible for the entering charge to fill the whole of this void completely and without delay on account of the fact that the velocity of outward movement of the mass of burnt gases is much higher than that of the entering charge, which enters the cylinder by expansion. This fact is illustrated by the exhaust pressure curves in Figures 11 to 14. These curves were all taken at the same engine speed, but with an admission pressure which is greater than atmospheric pressure by 100, 200, 300 and 400 gms. per cm², in Figures 11, 12, 13 and 14 respectively.

It will be seen from these curves that the well defined drop in pressure which is formed in the exhaust duct in the middle of the admission period still exists when the charge enters at a pressure of 400 gms. above atmospheric pressure.

In general this dip in the curves appears for many causes, which have the same result, which is an insufficiency of entering air from the admission to fill the void left by the exhaust gases in the cylinder and in the ducts traversed by the exhaust gases and left void. In certain cases this unfilled void may act in an objectionable manner upon the final charge in the cylinder by leaving it depressed below the pressure existing in the atmosphere external to the cylinder around the admission ports.

In practice, it is frequently difficult or incompatible for other reasons to make the admission orifices sufficiently large to permit the void to be filled. Further, the exhaust orifices may be too small to permit a sufficient charge to pass through the cylinder into the exhaust duct to fill the void or the exhaust orifices may be too large relative to the admission orifices so that the charge is absorbed by escapement through the exhaust orifices more rapidly than it can be admitted through the admission orifices.

We have seen that the absolute time of duration of the admission both as regards the fresh charge that has penetrated through the inlet ports and from the point of view of the return which limits the existence of the void is a factor which is not tied directly to the angular speed of rotation of the engine. These are times which have a constant duration in their absolute value with small variations due to other causes enumerated.

In order to permit a full utilization to be made of the void left in the cylinder and in the exhaust pipe by the mass exit of the burnt gases and to obtain the advantage of cooling the cylinder by the air that passes through the latter and enters the exhaust duct while avoiding any objectionable effect due to the fact that the void left in the exhaust duct behind the burnt gases cannot be completely filled, it is more rational to close the exhaust at a suitably chosen moment for the design of the engine. For example, the condition may be imposed that exhaust will be left open during the period of exit of the burnt gases and during a half, quarter or three quarters of the period of admission. During this time the cylinder is traversed by a current which cools the piston and the walls of the cylinder internally. If the exhaust is then closed, the current is stopped and no further passage of the charge into the exhaust duct can occur.

For example, the closure of exhaust may be produced after a constant lapse of time. The commencement of this time will be for example the opening of the exhaust orifices or any point upon the crank circle of the engine. In this case the time of passage of the current through the cylinder will always be constant in its duration.

Or a point may be chosen for the closure of exhaust upon the crank angle so that during the period between the opening and closure of exhaust the admission is produced and the current passes through the cylinder, in such a way that the closure of exhaust will remain always in the period of admission, so that with the variation in engine speeds the return impact will never come before this point and the end of the evacuation of the cylinder will never occur after this point.

This solution can be arrived at by all kinds of distributions such as the piston of the engine itself, sleeves, sliding members, rotary valves, baffles and other mechanical arrangements controlled by the rotation of the engine.

It is to be noted that these two solutions are easy to establish mechanically for the reasons that exhaust devices may be established which give sufficiently long durations of time between the end of the exhaust of the burnt gases from the cylinder and the return impact which reverses the current in the cylinder and the ducts.

Such methods are described in one or the other of applications Serial Nos. 738,014, 738,015, 738,016, 745,814, 38,826, 46,304, 46,805, filed August 1, 1934; August 1, 1934; August 1, 1934; September 27, 1934; August 31, 1935; October 25, 1935; October 25, 1935, respectively, and they concern the length of pipes, their shape, their volume, for example the shape of the tube which allows the gases to travel the furthest distance from the cylinder and gives the least violent return impact will be that of a tube which will have a cross section slightly greater than the surface of the exhaust orifices which is truly active during the exhaust period itself and which will have a slight conicity as the distance from the cylinder increases. The truly effective length is equal to the length of the path travelled by the column of exhaust gases as far as the point from which they rebound and this will be for the strongest charge in the cylinder in air and fuel, because this distance is proportional to the energy of the volume introduced, as explained in application Serial No. 84,182, filed June 8, 1936.

With the timing of exhaust closure as described above, it will be seen that over the chosen speed range of the engine, the exhaust will always close after the entering charge has had time to occupy the cylinder and enter the exhaust duct, before the return of the burnt gases can occur and before a prolonged suction can exert any action in drawing the charge out of the cylinder into the exhaust duct.

When the exhaust orifice is closed, however, the charge contained in the cylinder may be under a depression and it will therefore be advantageous for the admission to close later than the exhaust so that the balance of pressure be-

tween the cylinder and the external source of the fresh charge may be re-established.

It is to be observed that in the two cases indicated above and in which admission closes after the exhaust, the admission and the charging of the cylinder will be produced in two phases. In the first phase, fresh gases under depression will pass into the cylinder and into the exhaust duct and in the second phase when the exhaust has been closed the balance between the pressure in the cylinder and the pressure of the source of supply will be re-established.

The moment of opening of exhaust orifices and the moment of closure of the admission orifices must be chosen in such a way that the active cylinder volume is as large as possible, that is to say the volume contained in the cylinder at the moment when the last port closes.

The closure of exhaust before the closure of inlet established in the manner specified above is very advantageous in this connection.

Figure 10 shows by way of example a suitable port area diagram and timing of the inlet and exhaust events for a cylinder of 1.5 litres capacity, based on the foregoing remarks.

In this figure the curve 1 relates to the opening of the exhaust orifice and the curve 2 to the opening of the inlet orifice. The ordinates of the curves represent port areas and the abscissae represent crank angles.

In this figure the shaded portion *e* of the exhaust curve represents the period during which the mass exit of the burnt gases will occur over a wide range of engine speeds to ensure that inlet will always open as close as possible to the moment of mass exit of the burnt gases.

The overlapping portions of the two curves represent the period during which both the exhaust orifice and the inlet orifice are open and during which the charge will enter due to the difference in pressure between the source of supply and the interior of the cylinder and exhaust system, and it will be understood that the velocity acquired by the incoming charge will assist the charging and that this velocity will be highest when the void left by the burnt gases is greatest and the delay before the return of the burnt gases occurs is a maximum.

The portion *a* of the inlet curve represents the period during which the exhaust orifice is closed and the inlet orifice remains open for completing the charging and it will be understood that during this period the pressure in the cylinder will be restored to the pressure of the source of supply and that the velocity acquired by the entering charge will also assist the charging during this period *a*.

For this reason it is again advantageous that the delay in the return of the burnt gases should be a maximum.

I claim:

1. Method of controlling two stroke cycle internal combustion engines, which comprises establishing communication between the cylinder and exhaust system during the firing stroke, providing for the issuance of the burnt gases from the cylinder substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gases moves outward and thereafter returns from a point which may be within the exhaust system, providing a permanent free passage for the burnt gases to the limit of outward travel of said burnt gases, providing for the said issuance of the burnt gases to occupy a substantially constant crank angle over a chosen range of engine speeds, preventing the entrance of fresh charging air until the said issuance of the burnt gases is in full progress, admitting fresh charging air into the cylinder when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, while the exhaust port is still open, and providing for the said fresh charge to occupy the cylinder and a portion of the exhaust system in the interval elapsing between the said exit of the burnt gases and the instant when the pressure of the returning gases becomes effective within the cylinder.

2. Method of controlling two-stroke cycle internal combustion engines, which comprises establishing communication between the cylinder and exhaust system during the firing stroke, providing for the issuance of the burnt gases from the cylinder substantially as a mass in an interval of time shorter than that which would be required for the burnt gases to expand down to the ambient pressure by adiabatic flow, whereby the mass of gases moves outward and thereafter returns from a point which may be within the exhaust system, providing a permanent free passage for the burnt gases to the limit of outward travel of said burnt gases, providing for the said issuance of the burnt gases to occupy a progressively and substantial decreasing interval of time with increasing engine speed over a chosen range of engine speeds, preventing the entrance of fresh charging air until the said issuance of the burnt gases is in full progress, admitting fresh charging air into the cylinder when the said issuance of the burnt gases is in full progress and causes a suction effect to be exerted in the cylinder, while the exhaust port is still open, and providing for the said fresh charge to occupy the cylinder and a portion of the exhaust system in the interval elapsing between the said exit of the burnt gases and the instant when the pressure of the returning gases becomes effective within the cylinder.

3. A two-stroke cycle internal combustion engine having a cylinder, a piston moving in the cylinder, exhaust and inlet orifices in the cylinder, an exhaust conduit on the exhaust orifice, means for so controlling the exhaust orifice during the firing stroke as to ensure the issuance of the burnt gases as a mass, whereby the said mass moves outward and thereafter returns from a point which may be within the said conduit, means whereby the area of exhaust orifice opened per degree of crank shaft movement increases progressively substantially throughout the exhaust orifice opening period, means for so controlling the inlet orifice as to ensure that it will be opened while the exhaust orifice is still open and when the said issuance of the burnt gases is in full progress and produces a suction effect in the cylinder, the said conduit providing a permanent free passage for the burnt gases to the limit of outward movement of said gases and providing a passage for the gases during their outward motion as a mass having no cross section of substantially greater area than any cross section thereof further from the cylinder and no abrupt and substantial increase in volume for a length substantially equal to the limit of outward travel of the mass of burnt gases.

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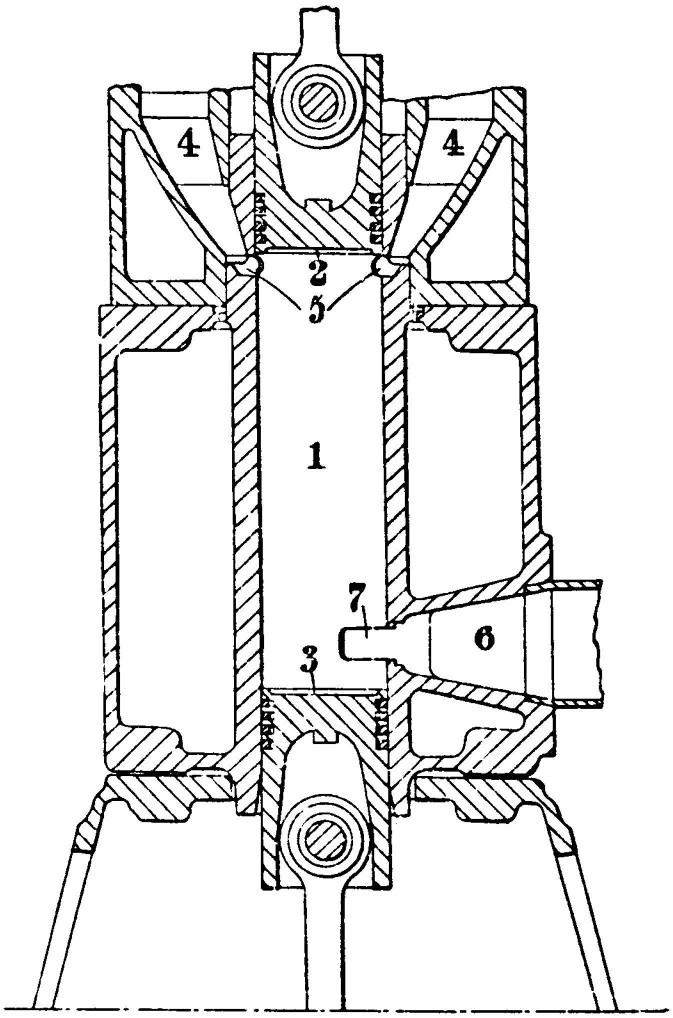

Fig.1.

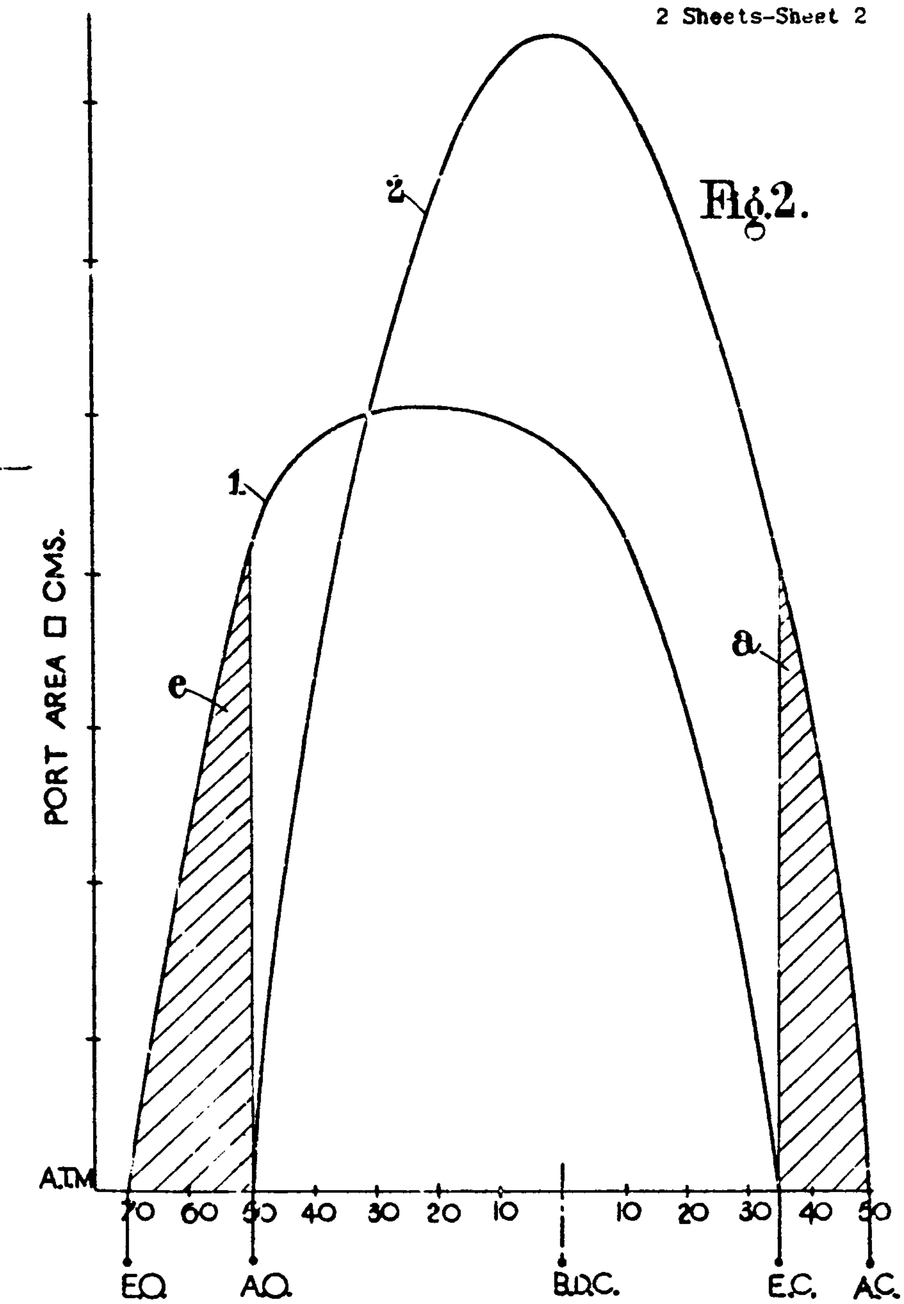

Fig.2.

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Patented Jan. 17, 1939 2,144,065

UNITED STATES PATENT OFFICE

2,144,065

INTERNAL COMBUSTION ENGINE

Michel Kadenacy, Paris, France, assignor of one-half to Armstrong Whitworth Securities Company, Limited, London, England

Application September 4, 1936, Serial No. 99,487
In Great Britain September 6, 1935

3 Claims. (Cl. 123—51)

This invention relates to two-stroke cycle internal combustion engines of the kind wherein at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when an adiabatic flow only is involved, and in such a short interval of time that it is discharged as a mass leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder by opening the inlet orifice with the required delay after the opening of the exhaust orifice to ensure that the burnt gases are then moving outwardly through the exhaust orifice or duct and that a suction effect is exerted at the inlet orifice as a consequence of the exit of the said mass.

In constructing or modifying a two-stroke cycle internal combustion engine of the kind set out above according to the present invention the relationship between the effective area of the exhaust lead, i. e. the effective area of exhaust orifice opened before inlet opens, and the angle of the exhaust lead, i. e. the crank angle between exhaust and inlet opening is arranged to conform with the following equation:—

$$\frac{W}{100\,KAv}=\frac{a}{360\,N}=t$$

where W is the cylinder volume in cm³, A the area of exhaust lead in cm², v a hypothetical mean velocity of mass exit of the burnt gases of the order of 450 metres per second (it should be understood that this is not an actual value), K a constant depending upon the form of the exhaust orifice and the area opened per unit movement of the piston or crank shaft (in other words allowance must be made for the gradual opening of the exhaust ports and for variation in the rate of opening of the ports in that the rate of opening of the ports varies with the method of actuation used), a is the angle in degrees of exhaust lead, N the speed of the engine in revolutions per second, and t is a time interval such as to ensure that the mass exit of the burnt gases from the cylinder will be completed in the interval elapsing between exhaust and inlet opening.

In the accompanying drawings:—

Figure 1 is an example of an engine to which the present invention may be applied, and

Figure 2 is an exhaust and inlet port area diagram for a cylinder of 1.5 litres capacity.

In Figure 2 the curve 1 relates to the opening of the exhaust orifice and the curve 2 the opening of the inlet orifice. The ordinates of the curves represent port areas and the abscissae crank angles.

EO and EC indicate the moments of exhaust opening and closure respectively and AO and AC the moments of inlet opening and closure respectively. BDC indicates bottom dead centre.

The above equation has been derived from the following considerations:

The total time occupied for the mass exit of the burnt gases to occur will be influenced by the area of the exhaust orifice available, see portion e of the exhaust curve, for example, since this area will determine the length of the column formed by the burnt gases in leaving the cylinder. This area of exhaust orifice is not opened instantaneously but progressively, so that the mean area available should be considered.

If the abovementioned column is too long, in other words if the orifice is too small, the time occupied by the mass exit of the burnt gases will be too great and this exit may not be completed in the time available before the inlet is or can be opened, or the expansion of the rear portion of the mass of burnt gases will become a dominant factor.

These factors must be taken into consideration in constructing an engine having a suitable timing of inlet opening, or in providing a suitable timing for an existing engine, whether this timing be obtained by altering the existing timing of the engine or by altering the area and/or rate of opening of the exhaust orifice, or by a combination of these possible alterations.

Over the chosen speed range it is necessary to ensure that the time elapsing before the inlet opens and the area of exhaust orifice available in this time will suffice to ensure that the mass exit of the burnt gases will be completed during this interval.

The time element which will suffice for this purpose will enable the crank angle between inlet and exhaust opening and also the area of exhaust orifice effectively opened in this crank angle to be established, and if this time requirement is satisfied for the highest engine speed, this will ensure that it will be satisfied at all lower engine speeds, i. e., over the whole or the chosen working speed range.

If either the area of exhaust lead or the angle of exhaust lead is chosen or is fixed and the mean speed of mass exit can be assumed with practical approximation, then the other of these two factors can also be determined so as to ensure that the desired object will be attained.

The inventor has found that calculations of sufficient accuracy to ensure practical results may be made by assuming that the cylinder vol-

ume of burnt gases is discharged, without expansion, at a hypothetical mean speed. This hypothetical mean speed of discharge will vary according to the fuel employed, the mixture and the conditions of combustion, among other factors. For fuel oil with good combustion a hypothetical mean speed of 450 metres/sec. may be taken although this hypothetical speed may be as low as 300 metres/sec. or as high as 600 metres/sec.

The length of the column formed by the passage of the mass of burnt gases through the exhaust orifice will be

$$\frac{W}{KA \times 100} \text{ metres}$$

For practical purposes it may be taken that $K = \frac{1}{2}$.

The time occupied by this mass exit will be

$$\frac{W}{100\,KAv} \text{ secs.}$$

The time elapsing between exhaust opening and inlet opening will be

$$\frac{a}{360} \times \frac{1}{N} \text{ secs.}$$

so that to satisfy the conditions of the invention, the following relationship should exist,

$$\frac{a}{360\,N} = \frac{W}{100\,KAv}$$

This will provide a relationship between A and a.

For practical purposes the two time elements equated above must suffice to ensure that the mass exit of the burnt gases occurs, but must not be too long. Further the angle a must be as short as possible in order to permit a suitable utilization of the crank angle available for charging.

For this time element the value .002 secs. for example may be taken as a basis for an engine of 1 litre capacity, and a speed of 1500 R. P. M., this value ensuring that the increased time interval at lower engine speeds is satisfactory for the purpose of the invention. Factors of correction should be introduced into this value for very large or very small cylinder volumes.

By this it is to be understood that this time will suffice to ensure that a mass evacuation of the cylinder occurs, but that this time may be increased or reduced if it is convenient. If this time element t is fixed, this will also fix the exhaust lead required at any engine speed and also the maximum lead required for the highest engine speed; and it will also fix the exhaust area required.

Since $$a = 360\,Nt$$

and $$A = \frac{W}{100\,Ktv}$$

in practice it will be found convenient to base the calculations on an angular interval of 25° between exhaust and inlet opening for an engine speed of 25 revs./sec. with a hypothetical mean speed of mass exit of the burnt gases of 450 meters/sec.

By way of example in a 3-cylinder opposed piston engine constructed by the applicant, the volume of each cylinder was 1220 ccs., the area of exhaust lead 19.66 cm², and the angle of exhaust lead 20°.

A single cylinder of such an engine is illustrated in Figure 1 of the drawings, which shows a cylinder bore 1, open at both ends, with two pistons 2, 3 working in opposition in this bore, each controlling respectively inlet and exhaust ports 8, 7 located at opposite ends of the cylinder and leading to ducts 4, 6 respectively, the individual pistons being operated by separate crank shafts or by a single crank shaft (not shown).

Assume a hypothetical mean speed of mass exit of the burnt gases of 450 metres/sec., and that $K = \frac{1}{2}$.

The length of the column of exhaust gas will be 1.25 metres.

The duration of time of mass exit will be $\frac{1}{360}$ secs. This will be the time the crank rotates through 20°. In other words the maximum engine speed that will be attained with the required timing of inlet opening will be 1200 R. P. M.

It was desired with this engine to have the required timing of inlet opening at all speeds up to 2400 R. P. M.

Based on the above data, this would require the area of exhaust lead to be increased to 39.32 cm.², with a lead of 20°.

It was found in practice that these figures were amply sufficient and that with an area of exhaust lead of 30 cm.² and a lead of 22° the desired result was still obtained, these figures showing that the hypoethetical mean speed of mass exit should have been in the neighbourhood of 500 metres/sec.

It should be noted that by means of the invention the result is ensured that inlet opens after exhaust only with the required delay to ensure that the burnt gases are moving outwardly through the exhaust orifice or system as a consequence of their mass exit from the cylinder. At any engine speed the earliest moment at which inlet can be opened in order to obtain advantage from the evacuation of the cylinder by the mass exit of the burnt gases will be the moment when the rear end of the mass of burnt gases, during its outward motion as a consequence of its mass exit from the cylinder, has passed the point at which the inlet orifice is situated and causes a suction into the cylinder at this inlet.

It will be appreciated that, other things being equal, as the engine speed increases, the moment at which inlet opens becomes situated more and more closely to the earliest moment defined above, until a maximum speed is reached when inlet opens at this earliest moment.

Two objectionable actions may arise which will cause the operation of the engine to be defective:

(1) At certain speeds disturbances are produced by the return of the burnt gases, which mixes the fresh air with the burnt gases and forces the charge out of the cylinder.

(2) At higher speeds the depression extends over a larger crank angle and as this depression also exists in the exhaust pipe it draws on the charge admitted into the cylinder and thereby reduces the weight of air in the final charge; consequently by closing exhaust before the return occurs at low engine speeds, the successful operation of the engine at higher speeds will be ensured.

This result may be obtained in the opposed piston engine referred to above, in which one piston controls the inlet ports and the other controls the exhaust ports by causing the crank of one piston to lead that of the other piston by a suitable amount.

Further, it is of great advantage to have the inlet and exhaust ports opened at the highest velocities, as in this way the rapid exit of the burnt gases and the inflow of the fresh charge will be facilitated.

In this engine by suitably offsetting the cranks with respect to the cylinder, the pistons may be caused to open the exit and inlet ports at the highest linear speeds.

In certain cases it may occur that the return of the burnt gases at low speeds takes place too soon to permit the engine to be charged or to permit a suitable and convenient closure of exhaust before the return of the burnt gases occurs, and in this case the engine will be modified in order to cause the return to occur sufficiently late for the purpose required.

This result may be obtained for example by utilizing means such as those described in my copending application No. 738,016 (Patent No. 2,110,986) for delaying the return of the gases; or by flaring the exhaust pipe outwardly, or within limits by lengthening the exhaust pipe. This result may also be obtained by arranging that the exhaust takes place at a moment when a depression exists or is produced in the exhaust pipe. For example in a multi-cylinder engine the exhaust from the individual cylinders may be so connected together that a cylinder exhaust into a pipe in which a depression has been previously produced by the exhaust from another cylinder and before this depression has been destroyed by the return of the burnt gases.

It should be noted that the invention is applicable to engines both in cases when the charge is admitted to the engine directly from the atmosphere and when it is introduced by a compressor.

The sole difference between these two cases consists in the difference in admission pressure at which the air enters the cylinder. Thus, for example, if we have an engine which is established in such a way that at the moment of opening the admission, the depression in the cylinder is 7 metres of water for example, the atmospheric air is pushed into the cylinder with a force of 700 gms. per sq. c. m. If a compressor is connected on the engine and this compressor gives a pressure of 300 gms. per sq. c. m., then in this case the air is pushed into the cylinder with a pressure of 700 gms., plus 300 gms., making a total of 1 Kg. Consequently it will be advantageous to provide compressors of the centrifugal type for example. Air can pass freely through these compressors, independently of the rotation of the rotor and consequently the first and massive entry of air into the cylinder will not be restricted as in a volumetric compressor.

The advantage of this type of compressor is that it permits the engine to work by direct admission through the compressor, even when the latter has stopped and is not working.

Any compressor so used could be arranged so that the air normally passes freely to the engine from the atmosphere, the compressor then being actuated when required in order to increase the charge passing through the compressor or for supercharging such as is necessary for example on an aircraft engine.

What I claim is:—

1. A two-stroke cycle internal combustion engine of the kind set forth wherein the area of exhaust lead and the angle of exhaust lead conform with the following equation:—

$$\frac{W}{100\,KAv}=\frac{a}{360\,N}=t$$

where W is the cylinder volume in cm.³, A the area of exhaust lead in cm.², v a hypothetical mean velocity of mass exit of the burnt gases of the order of 450 metres per second, K a constant depending upon the form of the exhaust orifice and the area opened per unit movement of the piston or crank shaft, a is the angle of exhaust lead in degrees and N the speed of the engine in revolutions per second, the time t being such as to ensure that the mass exit of the burnt gases from the cylinder will be completed in the interval elapsing between exhaust and inlet opening.

2. A two-stroke cycle internal combustion engine as claimed in claim 1, wherein a equals 20° where N equals 25 revolutions per second.

3. A two-stroke cycle internal combustion engine of one litre capacity as claimed in claim 1, wherein t is equal to .002 secs.

MICHEL KADENACY.

July 12, 1938. M. KADENACY 2,123,569

TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed Jan. 11, 1937 3 Sheets-Sheet 1

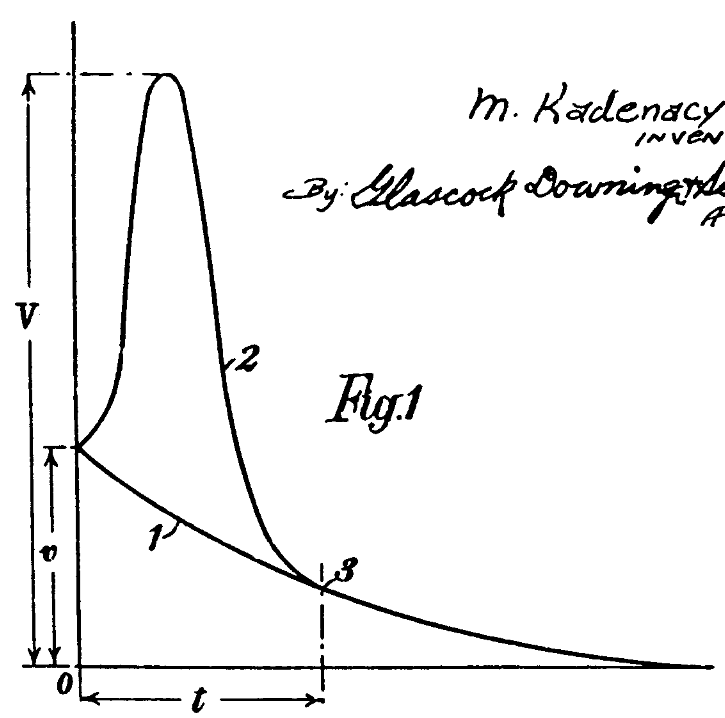

Fig.1

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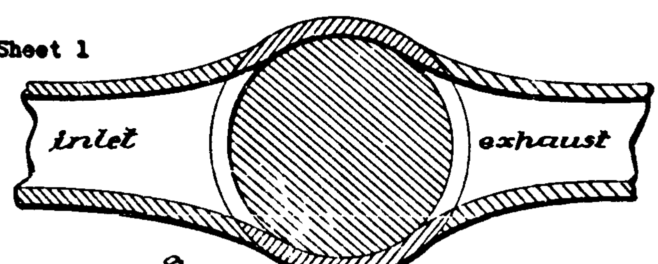


Fig.5.

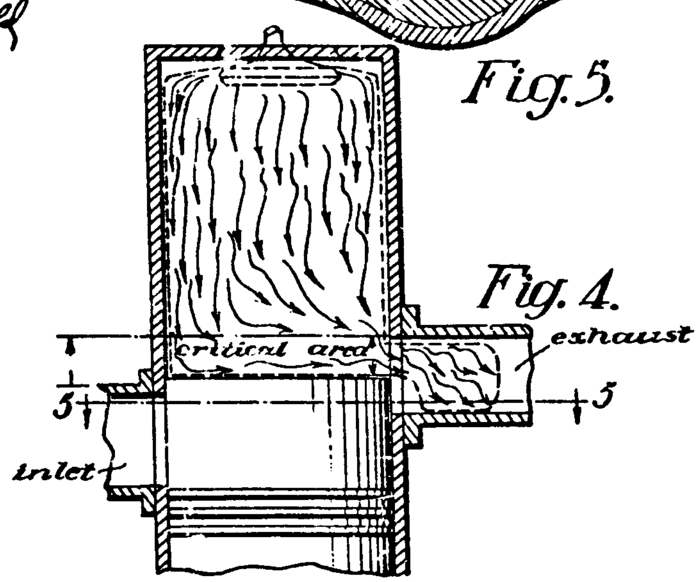
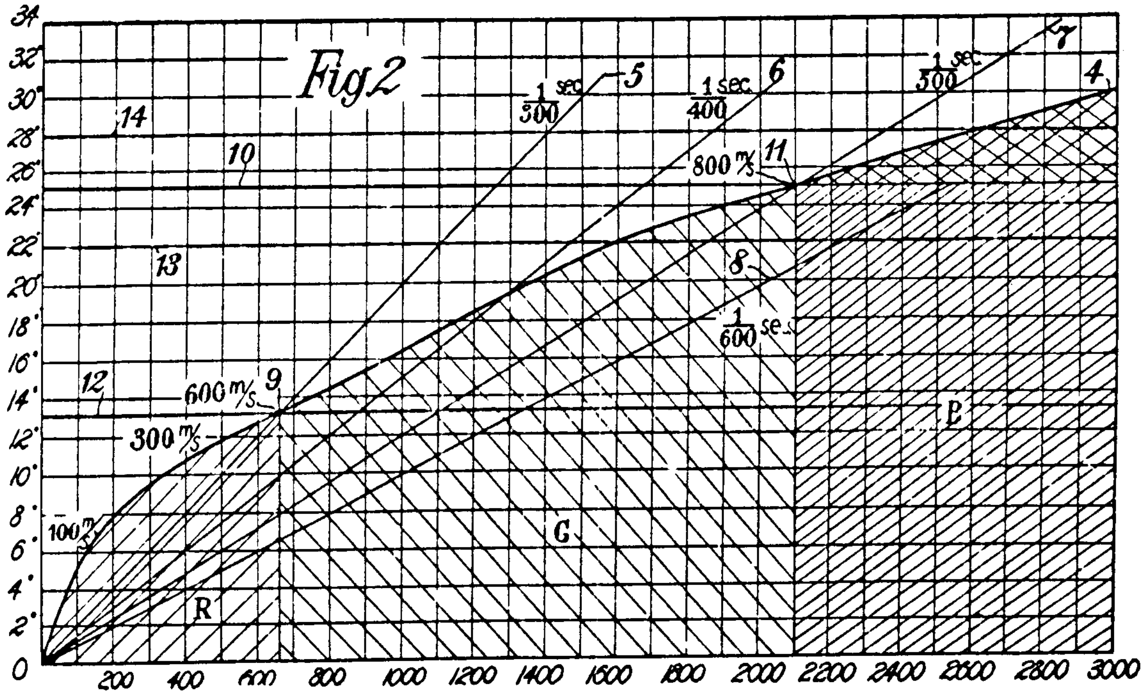


Fig. 4.

July 12, 1938. **M. KADENACY** **2,123,569**

TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed Jan. 11, 1937 3 Sheets-Sheet 2



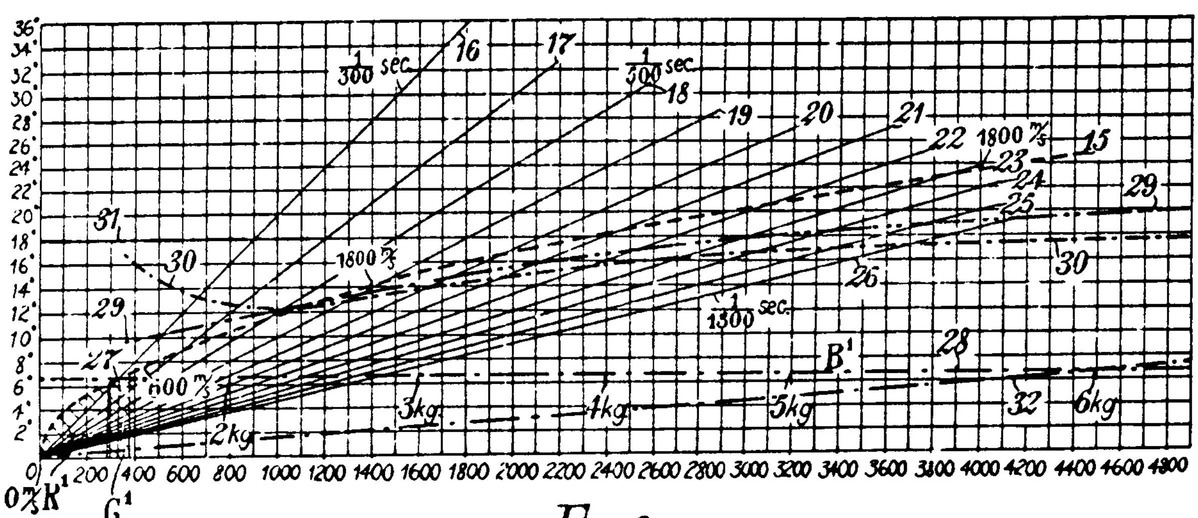


Fig.3

2,123,569

3 Sheets-Sheet 3

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Patented July 12, 1938 2,123,569

UNITED STATES PATENT OFFICE

2,123,569

TWO-STROKE INTERNAL COMBUSTION ENGINE

Michel Kadenacy, Paris, France, assignor of one-half to Armstrong Whitworth Securities Company, Limited, Westminster, London, England

Application January 11, 1937, Serial No. 120,118
In Great Britain January 11, 1936

1 Claim. (Cl. 123—65)

This invention relates to two-stroke cycle internal combustion engines which operate in conformity with the natural phenomena accompanying the combustion and discharge of the gases, that is to say, to engines in which the high depression or vacuum occasioned as a consequence of the discharge of the burnt gases from the cylinder into the exhaust system as a coherent mass at a speed in excess of the speed of adiabatic expansion, by virtue of the energy contained in the gases while still in the cylinder, is utilized in charging the engine.

More particularly the invention relates to two-stroke cycle internal combustion engines, wherein the evacuation of the cylinder by the mass exit of the burnt gases is utilized in introducing the fresh charge by opening the inlet when the burnt gases are moving outwardly through the exhaust port or duct as a consequence of their mass exit from the cylinder.

The object of the invention is to provide a method of constructing such an engine, whereby an optimum and stable torque may be obtained, and additionally a constant torque over a desired speed range.

The invention consists in arranging that the area of exhaust orifice opened, within a sufficiently short interval of time to ensure that the gases then still in the cylinder retain sufficient ballistic energy to maintain a speed greater than the speed of adiabatic expansion during the subsequent evacuation of the cylinder, is greater than the product of the area of the engine piston multiplied by the instantaneous speed of adiabatic expansion for the medium considered and divided by the instantaneous speed of ballistic exit.

The area of exhaust defined in the above paragraph will hereinafter be referred to as the critical area of exhaust.

The invention further consists in arranging for an additional area of exhaust orifice to open after the said critical area has opened in order to ensure a more rapid evacuation of the burnt gases.

The invention still further consists in arranging for the said critical area to be opened within the required interval of time at the lowest engine speed.

The invention will be more clearly understood from the following description and by reference to the accompanying drawings, in which:

Figure 1 is an explanatory diagram relating to the velocity of exit of the burnt gases upon their discharge from the cylinder through the exhaust orifice.

Figure 2 is a diagram illustrating the different conditions obtained in the cylinder of an internal combustion engine as a consequence of the discharge of the burnt gases and showing the effect of a retarded opening of the critical area.

Figure 3 is a similar diagram to Figure 2, but showing the effect of an earlier opening of the critical area.

Fig. 4 is a central section through the cylinder showing the inlet and exhaust.

Fig. 5 is a section at right angles through Fig. 4 on the line 5—5.

It has been the common belief that when the burnt gases are discharged from the cylinder of an internal combustion engine, the discharge of the burnt gases is in the nature of an adiabatic expansion and the speed of discharge of these gases is the speed of sound for the medium considered.

The applicant has already indicated in prior specifications that this assumption is not in accordance with fact, and that the behaviour of the burnt gases upon and after their discharge from the cylinder is such as to lead to the belief that the burnt gases, while still in the cylinder, form a body having properties similar to those of a resilient body, and which upon the opening of the exhaust orifice, seeks to project itself as a coherent mass from the cylinder.

They have observed that when the exhaust orifice opens there is first a period of delay, during which no appreciable change occurs in the gaseous medium external to the exhaust orifice and that after this delay has elapsed the burnt gases issue from the cylinder at a speed greatly in excess of the previously assumed speed of adiabatic expansion and as a coherent mass, the motion of which is governed by the laws of reflection and rebound.

This is in no way to be understood as excluding the expansion of the gases during their discharge from the cylinder. During this discharge the adiabatic expansion of the burnt gases occurs continuously but, on account of the fact that their speed of exit is greatly in excess of the speed of sound, the gases display consequent properties of cohesion so that they are capable of separating themselves from the walls of their containing vessel.

In order to give a clearer understanding of the phenomena under consideration and of the present invention, an analogy may be drawn by

referring to the behaviour of a coil spring in reacting to impressed forces.

If a free helically coiled spring is placed on a table and is subjected to compression, the force of compression stored in the spring is a force acting on a mass and capable of imparting momentum to this mass. If the spring is then released so that it expands against a resistance, and the resistance is suitably chosen, the spring can be caused to expand slowly until it returns to its free length in which it will remain in a state of rest.

The work done by the spring in expanding will then have become stored in the resistance.

On the other hand, if the spring, after having been compressed on the table, is suddenly released, that is to say the compressing means are removed in a very small interval of time, the spring, while expanding as a consequence of its release, will also leave the table bodily. The energy stored in the spring has been utilized wholly in imparting momentum to the spring. During its flight through the air after it has left the table, oscillations will occur in the continuous medium constituting the spring, but these oscillations will bear no direct relationship with the motion of the spring bodily from the table.

This condition may be compared with the exit of the burnt gases from the cylinder at a speed higher than the speed of adiabatic expansion.

Again, if while the spring is held in a static state of compression between two surfaces it is subjected to an impact, for example a hammer blow, the action exerted on the spring by the hammer blow will manifest itself in two forms. In the first place it will increase the compression of the spring and in the second place it will cause motion of the centre of gravity of the spring, since it is a force acting on a mass and capable of producing an acceleration of this mass.

If one of the retaining walls is then removed in such a short interval of time that this motion of the centre of gravity of the spring still continues, the spring will rebound bodily from the other wall with a speed which is increased as a consequence of the velocity imparted to its centre of gravity by the hammer blow.

The hammer blow imparted to the spring may be compared with the impact exerted on the compressed gases in an engine cylinder by the combustion of these gases, and it will be seen that the suggestion that is being made is that this combustion, in addition to increasing the pressure of the gases, may also impart velocity to the centre of gravity of the mass of gases in the cylinder, and that the speed of exit of the burnt gases when the exhaust orifice is opened is influenced by this possible pre-existing velocity in the cylinder.

From the above explanation it will be seen that in the applicant's view the burnt gases which are discharged from the cylinder of an internal combustion engine have a mechanical elasticity which can be compared with that of a spring and that it is the time element that determines whether the gases will leave the cylinder by a flow set up by adiabatic expansion or as a body projected by applied force.

The adiabatic expansion of gases may be compared with the condition under which the expansion of the spring is exerted upon an opposed resistance. The mass exit of the burnt gases from the cylinder may be compared with the condition under which the major part of the energy stored in the spring is transformed into motion of the mass of the spring.

When the burnt gases leave the cylinder through the exhaust orifice, their velocity, since it is greatly in excess of the speed of sound for the medium considered, may be considered to be the consequence of two factors,

(1) an adiabatic flow having a velocity equal to the speed of sound, and

(2) a force capable of imparting to the gases a velocity greatly in excess of the speed of sound.

This force will hereinafter be referred to as the ballistic force and the velocity which is a consequence of this force will be called the ballistic speed of the burnt gases.

Under the action of the first of these factors the gases are only capable of expanding as a continuous medium maintaining contact with all parts of the walls of the containing vessel, and until a pressure equilibrium is reached between the cylinder and the external atmosphere.

Under the action of the second of these factors, since the centre of gravity of the mass of burnt gases is moved at a speed in excess of the speed of sound, the mass of burnt gases is capable of separating itself from its containing walls.

In considering the analogy between the burnt gases in an engine cylinder and the compressed spring, it must however be borne in mind that in the engine the burnt gases are discharged from the cylinder through an orifice which is always smaller than the area of cross section of the cylinder and which is opened gradually and not instantaneously. The release of the gaseous spring formed by the burnt gases is thereby resisted and the greater this resistance the less will be the energy that is transformed into momentum of the mass of the gaseous spring.

If we consider that the burnt gases have arrived in the exhaust duct and are passing outwardly through the latter at their instantaneous ballistic speed, and neglect expansion, then on account of the difference in area between the exhaust orifice and the area of the engine cylinder, the centre of gravity of the portion of the burnt gases remaining in the cylinder will adopt a resultant movement towards the outlet at a slower speed than that of the portion of the burnt gases contained in the exhaust duct and the ratio between these two speeds will be determined by the ratio between the area of exhaust orifice open at this moment and the area of cross section of the cylinder.

But since the whole mass of burnt gases is expanding continuously at its speed of adiabatic expansion, the speed of exit of the portion of the burnt gases contained in the cylinder at any moment will be reduced by the speed of this expansion.

Expressed in other words, the increment of volume by expansion occurring in the cylinder in any interval of time, is deducted from the effective reduction in volume of the cylinder content by escape through the exhaust orifice in this same interval of time.

It will therefore be seen that according to the predominance of one or the other of the two factors mentioned above, at the end of the discharge of the burnt gases as a consequence of both of these actions, the cylinder space will either be full of gases at atmospheric pressure in an inert state, or it will contain gases in a rarefied state, or this space will be completely void of burnt gases.

It is now proposed to analyze these three conditions with reference to Figure 1 of the accompanying drawings.

This figure shows two velocity curves, on a time base, the ordinates representing velocities.

The point 0 represents the moment at which the exhaust orifice commences to open.

The curve 1 represents the changes in velocity that occur in a gas which is expanding from a high pressure to a low pressure, and when a purely adiabatic flow has been established.

The ordinate v represents the maximum velocity which will be obtained when the pressure difference is greatest (that is to say, at the moment when the exhaust orifice opens), and it will be seen that this velocity falls gradually as the time interval increases, until it eventually reaches zero.

Such a curve may be established by calculations based on considerations well known in the art, and is an imaginary curve of what is assumed to occur when a gas is expanded from a high pressure to a low pressure.

The applicant has observed that the changes in velocity that occur initially as a consequence of the opening of an orifice for the release of the compressed burnt gases in an internal combustion engine are not those represented by the curve 1. On the contrary, with a certain delay the velocity of exit rises rapidly to a speed greatly in excess of the maximum velocity of adiabatic expansion. If the duration of exit is prolonged, this high speed of exit then falls and eventually becomes identified with the adiabatic speed.

This initial high or ballistic speed manifests itself as a consequence of the opening of an exhaust orifice, irrespective of the size of this orifice, but evidently the volume of gases discharged at this initial ballistic velocity, which exists for only a very short interval of time after the burnt gases have commenced to leave the cylinder, will depend upon the magnitude of the orifice.

These conditions are represented by the curve 2 in Figure 1.

It will be seen that in this case the velocity rises rapidly to a peak value V which is greater than the maximum velocity of the curve 1 and subsequently falls first rapidly and then at a decreasing rate until it finally merges with the curve 1 at the point 3 in a time t which will be hereinafter referred to as the critical time.

Following the analogy of the spring, it may be imagined that the curve 1 represents the variations in rate of expansion that occur when the compression of the spring is released so gradually that the reaction exerted by the table against which the spring is compressed is absorbed by doing work on another medium than on the spring, and no propelling action on the spring occurs. In the case of the curve 2 the higher ordinate V may be considered to be due to the propelling action arising either solely from the compression of the spring at this moment or from this action in combination with the preexisting velocity of the centre of gravity of the spring produced by the hammer blow, that is, by the combustion of the charge. In this case the gaseous spring upon its release will be subjected to an acceleration which is represented by the initial rise in the curve 2 and after a period of time t when the energy of propulsion is dissipated any further motion of the centre of gravity can only be produced by simple adiabatic expansion, the curve 2 will become tangential with the curve 1 at the point 3. In the curve 2 the initial velocity has been take as v but it may be higher or lower than this value since if the hammer blow is taken into consideration as producing a pre-existing velocity of the burnt gases in the cylinder, then at the moment the exhaust orifice opens, the burnt gases may be moving towards or away from the orifice at this preexisting high velocity.

It will be seen from the above that at the end of the interval of time t the velocity of the burnt gases which can be caused by all the factors that have any influence on the velocity of the gases has fallen to such a low level that it becomes equal to the velocity that can be produced solely by a simple adiabatic flow.

At this moment it may be said that the speed of exit of the burnt gases becomes the speed of adiabatic expansion of the burnt gases, which is effective in all directions, so that at any moment thereafter the burnt gases will completely fill the space in which they are accommodated. In an engine of the kind to which the invention relates the discharge of the whole or a substantial portion of the burnt gases occurs in that part of the curve 2 which lies to the left of the point 3.

If the burnt gases have left the cylinder of the engine through an orifice and in such a way that at the end of the time t the pressure in the cylinder is above atmospheric pressure, then thereafter the burnt gases will simply expand adiabatically until a pressure balance has been reached between the cylinder and the external ambient medium.

When an exhaust system is fitted the equalization of pressure will be established between the medium in the cylinder and the medium in the exhaust system, but the exhaust system is under the influence of the gases that have left ballistically and account must be taken of this action upon the interior of the cylinder in the following moments.

If, however, the exhaust orifice is opened and in a time interval shorter than t and at the ballistic speed of exit of the gases a pressure balance is reached, then thereafter, since the burnt gases are still propelled outwards at their ballistic speed, a depression will be created in the cylinder. This is a second case.

In this case also a controlling action may be obtained by the use of a suitably designed exhaust duct, whereby the depression in the cylinder may be intensified and its duration prolonged by the action of the burnt gases which have left the cylinder ballistically.

Further, if in an interval of time less than t, the area of exhaust orifice opened is greater than the area of the engine piston multiplied by the instantaneous speed of adiabatic expansion and divided by the instantaneous speed of ballistic exit, then, at this moment, the expansion of the tail end of the burnt gases contained in the cylinder will no longer be a predominant factor, and this tail end of the burnt gases will detach itself from that portion of the cylinder most remote from the exhaust orifice and a complete void will be left behind the burnt gases in the cylinder. This is a third case.

The shorter the interval of time in which this critical area is opened, the less the expansion of the burnt gases that will have occurred in this time interval, and consequently the more compact will be the tail end of the burnt gases, and the more ballistic energy will remain in the burnt gases, so that the volume of complete void that will be left behind the tail end of the burnt gases will be the greatest.

For the ballistic speed of exhaust a value of from 1400 to 1800 metres/sec. may be assumed, according to the quality of the combustion and the degree of compression. These figures relate to fuel oil, of the kind commonly employed in compression ignition engines and will vary with the type of fuel employed. For petrol, the figures will be somewhat higher.

For the speed of adiabatic expansion, figures of 350 to 450 metres/sec. may be assumed, so that if the higher figure is taken in each case, the critical area of exhaust orifice will be $^{450}/_{1800}$ or ¼ the cross sectional area of the engine cylinder.

For the critical time t the value $^{1}/_{300}$ sec. may be assumed, since the applicant has found that it is in the neighbourhood of this time interval that a very satisfactory utilization of the ballistic phenomena is obtained for the purpose of the present invention.

The manner in which the above considerations may be applied in the construction of an engine will appear from the following and with reference to Figures 2 and 3, but it must be understood throughout the present specification that where numerical values are assumed for the adiabatic or ballistic speeds of the burnt gases and for the critical area of exhaust and for the critical time interval referred to above, these values are practical values which have been obtained as a result of experience and that although they suffice for establishing an engine in accordance with the invention, they are not necessarily exact values.

As a first method of calculation it may be assumed that at the moment of opening of exhaust the gases in the cylinder are at a pressure of 5 atmospheres, and that 4 cylinder volumes of burnt gases must be discharged to restore an instantaneous and figurative pressure balance between the interior of the cylinder and the atmosphere external to the exhaust orifice, these 4 volumes being discharged at the ballistic speed of 1,800 metres/sec. On the basis of these considerations a curve can be constructed, the points on which denote the crank angle at which this pressure equilibrium is reached at varying engine speeds. On such a curve, when these 4 volumes have been discharged in a time less than the critical time t, the end of exhaust will occur at the ballistic speed of the burnt gases.

As an alternative method of approximate calculation, the applicant has found that in engines of the kind to which the invention relates, a hypothetical speed of 450 metres per second may be applied to the exit of one cylinder volume of burnt gases, neglecting expansion but including the delay period that elapses before the burnt gases commence to leave the cylinder. The time interval that will be required to discharge the burnt gases from the cylinder can be calculated from a consideration of the mean area of opening of the exhaust orifice.

Either of the above methods of calculation will give the same result, but the curve shown in Figure 2 is obtained by the second of these methods.

In this figure the ordinates represent degrees of crank movement starting from the opening of exhaust which is marked by the point 0 and the abscissae represent revolutions per minute of the engine.

The curve 04 indicates the angular interval at which the burnt gases have been discharged from the cylinder at the different speeds of revolution of the engine, in so far as the action within the cylinder is concerned. This curve is obtained as follows:

Assuming the volume of the cylinder to be represented by W

the angular interval between the opening of the exhaust and any chosen point on the crank angle by a

the mean area of exhaust orifice opened in this crank angle by A,

then the length of the column formed by the issuing mass of burnt gases will be

$$\frac{W}{A}$$

the time interval that will be occupied for the mass of burnt gases to travel this distance past the exhaust orifice at 450 metres/sec. will be

$$\frac{W}{A}\times\frac{1}{450}$$

It is then necessary to determine the engine speed at which in the chosen number of degrees of crank angle the mean area of exhaust orifice in question has been opened in the above mentioned time interval, and this will enable points on the curve to be obtained for each selected angular interval after exhaust opening has commenced.

Polar co-ordinates representing constant time intervals elapsing after the opening of exhaust at varying engine speeds are then drawn from the point 0, the lines 05, 06, 07, 08 representing respectively $^{1}/_{300}$ sec., $^{1}/_{400}$ sec., $^{1}/_{500}$ sec., $^{1}/_{600}$ sec.

If a vertical ordinate is drawn at the point 9 at which the curve 04 intersects the line 05, it will be seen that this ordinate, in the example chosen, corresponds with a speed of approximately 660 R. P. M.

At all speeds higher than 660 R. P. M., in the area marked G, it may be assumed that the burnt gases are discharged from the cylinder in an interval of time less than $^{1}/_{300}$ sec., and consequently that the cylinder is left in a rarefied condition or is completely void of gases.

At all speeds under 660 R. P. M., it will be seen that the burnt gases are discharged in an interval of time which is greater than $^{1}/_{300}$ sec., so that in the area below the curve 04 marked by the reference letter R, the end of the exhaust occurs at speeds progressively approaching those of adiabatic expansion only, leading eventually, at some unspecified lower speed to an equalization of pressures between the cylinder and the external atmosphere.

A horizontal line is then drawn in the diagram to represent the crank angle at which the critical area of exhaust orifice is opened. It will be seen that this line which is designated by the numeral 10 in the drawings intersects the curve 04 at the point 11 which corresponds with a speed of 2,100 R. P. M. At all higher speeds than 2,100 R. P. M., in the area marked B, the critical area of exhaust orifice has been opened in an interval of time sufficiently smaller than $^{1}/_{300}$ sec. to ensure that after this critical area has been opened the speed of the burnt gases then contained in the cylinder, in the direction of outlet, will be sufficiently greater than the speed of adiabatic expansion so that the tail end of the gases then in the cylinder will constitute what may be considered to be an imaginary gaseous piston which detaches itself from the end of the cylinder, leaving a complete void behind it.

It should be clearly understood that the tail end of the burnt gases at this moment may be in either a rarefied or compressed state and that this will depend upon the time interval.

At the point 11 the tail end of the burnt gases will be highly rarefied, but thereafter as the speed increases the imaginary gaseous piston will be more and more highly compressed, while still detaching itself from the cylinder head, since the interval of time that has elapsed before the critical area has opened becomes shorter, and the amount of expansion of the burnt gases that can have occurred is also smaller.

In addition as the speed of the engine increases after the point 11, the quantity of burnt gases that has left the cylinder at its ballistic speed before the critical area of exhaust is opened will be smaller so that the energy contained in the burnt gases remaining in the cylinder when the critical area of exhaust is opened will be greater.

In the engine under consideration it will be now assumed that inlet is timed to open at 13° after exhaust opens, the opening of inlet being represented by the horizontal line 12.

It will be seen that this line passes through the point 9 which is at the intersection of the curve 04 with the line 05 representing 1/300 sec.

In this engine, therefore, at speeds less than 660 R. P. M., at the end of the exhaust operation the residual gases in the cylinder have been left in a progressively less rarefied condition by the discharge of the burnt gases at their ballistic speed.

At speeds higher than 660 R. P. M., at the end of the exhaust operation the cylinder has been left in a progressively more highly rarefied condition by the discharge of the burnt gases at their ballistic speed, but at all these higher speeds of the engine, inlet is opened too soon to permit a good utilization of this depression or rarefaction.

If the timing of inlet opening is then altered in order to permit a good utilization of the depression left in the cylinder by the mass exit of the burnt gases as set forth in British specification No. 431,856 and in other specifications in the name of the present applicant, for example so that the interval between exhaust opening and inlet opening is 22° on the line 13, then at 1,600 R. P. M., the engine will be capable of operating by drawing its charge directly from the atmosphere and it will give its maximum torque at this speed although this will not be the optimum possible torque for this engine.

At higher speeds than 1,600 R. P. M., inlet will open too soon and the torque will fall off rapidly.

Below 1,600 R. P. M. the crank angle between the discharge of the burnt gases from the cylinder and the opening of inlet increases and in addition the depression left in the cylinder decreases so that eventually the torque again falls off.

If the angular interval between exhaust opening and inlet opening is further increased to 28° on the line 14, then at 2,600 R. P. M. the critical area of exhaust has been opened in an interval of time considerably smaller than 1/300 sec., and before the ballistic speed of the burnt gases has become non-existent. Consequently the imaginary gaseous piston will separate itself from the cylinder, and the cylinder will be completely evacuated and capable of receiving a full charge by atmospheric pressure under the best possible conditions.

It will then be seen that the range of working speeds at which the engine can operate with a fixed timing of inlet and by drawing its charge directly from the atmosphere has been considerably increased and that at its highest speed range the conditions of operation will be such as to permit an optimum torque to be obtained.

By way of example only, numerals indicating the velocity of the burnt gases have been placed at different points on the curve 04 in order to indicate the manner in which the velocity at the instantaneous end of exhaust increases as the interval of time occupied by the discharge of the burnt gases decreases.

It will be appreciated from a consideration of Figure 2 and the above description that the most desirable conditions for the engine will be those in which the critical area is opened in the smallest possible crank interval after the opening of exhaust and in which this critical area of exhaust orifice is opened within the critical time interval at the lowest engine speed.

Such a condition is illustrated in Figure 3.

This figure is similar to Figure 2 in respect of the showing of the curve 015 and the polar coordinates 016 to 026 representing constant time intervals.

In this example it is assumed that it is desired that at all speeds above 600 R. P. M. the cylinder shall be left completely void as a consequence of the discharge of the burnt gases at their ballistic speed, that is to say, that the imaginary gaseous piston will always separate itself from the cylinder head and will thereafter move continuously out of the cylinder.

In order to obtain such a condition, the engine is so arranged that the critical area of exhaust orifice is opened within the critical interval of time at a speed lower than 600 R. P. M. In the example the critical area is opened within an interval of 1/300 sec. at 350 R. P. M., the critical area being opened in 7° of crank movement after the opening of the exhaust. This pre-supposes a larger periphery of the exhaust orifice and has the effect of flattening the curve 015.

This will have the effect of reducing the range of speeds over which the area G of the curve (Figure 2) extends and in which the cylinder is left in a rarefied condition, but not completely void.

The distance between the points 9 and 11 may in fact be so reduced by suitably situating the critical area of exhaust orifice that the portion G of the curve, for all practical purposes, may be considered to be nonexistent, and so that the engine operates throughout its whole speed range under the best possible conditions, with the cylinder completely empty of burnt gases at the end of the discharge of the latter.

In Figure 3 the line 016 representing the critical time of 1/300 sec. intersects the curve 015 at the point 27 and the line 28 representing the critical area of exhaust orifice also intersects the curve 015 at the point 27.

In the portion R_1 of the curve 015 to the left of the point 27, the cylinder, at the end of the discharge of the burnt gases therefrom, and neglecting the action in the exhaust system of the portion of the gases which has left the cylinder ballistically, remains full of gases at atmospheric pressure while to the right of the point 27, in the part B_1, the imaginary gaseous piston always separates itself from the end of the cylinder. But for the sake of accuracy, since the intermediate condition cannot be made to disappear wholly, a narrow portion G_1 of the curve is shown overlapping the point 27.

It should be understood clearly that the curve 015 is an imaginary curve, and gives no indication of where the burnt gases are situated at the assumed end of the discharge, or when the four cylinder volumes have been discharged.

The portion of the curve 615 which is of interest in establishing the timing of inlet opening, is that part which lies to the right of the point 27 on the $\frac{1}{300}$ sec. line, since it is in this part of the curve that the imaginary gaseous piston separates itself from the end of the cylinder.

As stated above, when the critical area of exhaust orifice has been opened, the imaginary gaseous piston will commence to separate itself from the cylinder head. At this moment a certain quantity of the burnt gases has already been discharged from the cylinder, but the volume of the cylinder remains fully occupied by burnt gases.

It may, therefore, be assumed that after this moment one cylinder volume of burnt gases has to be discharged through the exhaust orifice at a mean speed of 1,800 metres per second, the mean area of the exhaust orifice through which this exit occurs being measured from the moment when the critical area is opened until the moment when the tail end of the burnt gases leaves the cylinder.

In order to establish the moment when the evacuation of the cylinder is completed over this part of the curve 615 a further calculation may be applied in the following manner.

Such a calculation should take into consideration the expansion of the burnt gases during their ballistic exit from the cylinder, and the crank angles and mean areas of exhaust orifice to be considered must be measured from the crank angle at which the critical area of exhaust opens.

For each chosen crank angle after the opening of the critical area of exhaust, the mean area of exhaust open during the interval in question may be determined, and it is clear that these mean areas will always be greater than the critical area.

For practical purposes the mean speed of ballistic exit in the exhaust pipe may be assumed to be 1,800 metres/second, and the mean speed of adiabatic expansion 450 metres/second. If these values are taken, the calculation may be carried out as follows:

If A_p is the area of cross section of the cylinder

A_c is the critical area of exhaust orifice. (In the example and according to the invention this will be greater than

$$\left.\frac{A_p\times 450}{1800}\right)$$

A_x is the area of exhaust orifice opened at the chosen crank angle.

Then the mean area of exhaust orifice A_m equals $(A_x+A_c)\times$ a time/area factor say $\frac{1}{2}$.

The speed of outward movement of the volume of gases in the cylinder will be

$$\frac{A_m}{A_p}\times 1800$$

which will be greater than 450 metres/second.

Since the gases are expanding backwards at 450 metres/sec., the resultant speed of discharge of the cylinder contents will be

$$\left(\frac{A_m}{A_p}\cdot 1800\right)-450 \text{ metres/sec.}$$

From the length of the stroke the time interval occupied to evacuate the cylinder completely at this resultant speed of exit can be determined, and from this time interval the engine speed at which the mean area of exhaust in question has been opened can be determined.

A calculation carried out in this way will give a curve such as the curve 29 which denotes the moments when the cylinder has been left completely void.

It will be seen that at the lower speeds this curve becomes situated above the curve 615 and that at higher speeds it becomes situated below this curve.

This curve 29 is subject to correction since it is established on the assumption of a constant ballistic speed of exit of the burnt gases after the critical area has been opened and a constant speed of adiabatic expansion.

If it had been derived from the instantaneous values of these speeds the curve would have adopted the form of the curve 30.

In this connection it should be observed that at low engine speeds, say at 800 R. P. M., a greater proportion of the burnt gases has left the cylinder before the critical area of exhaust orifice has opened than at high speeds, say 4,000 R. P. M.

Consequently, at low speeds the smaller mass of the volume of burnt gases filling the cylinder at the moment the critical area of exhaust is opened will be less dense and will contain less ballistic energy. Its subsequent speed of exit will be lower, and it will travel a shorter distance from the cylinder after the total evacuation has occurred.

In the limiting case the gaseous piston will move a certain distance down the cylinder and will thereafter become stationary, and will remain in the cylinder. When this condition arises, it will be clearly understood that the gases in the cylinder are nevertheless in a highly rarefied state and expansion immediately becomes the dominant factor, resulting in refilling of the void with rarefied gases unless the inlet is opened at this moment.

At the higher speeds the volume of burnt gases filling the cylinder when the critical area of exhaust has opened is more dense and compact, the amount of ballistic energy retained in the cylinder at this moment is greater and the ballistic speed of exit of the remaining burnt gases is higher so that the time occupied for the subsequent evacuation of the cylinder will be smaller.

By way of example, velocities of exit of the burnt gases are indicated on the curve 615 which may be considered in conjunction with the above remarks in considering the form of the curves 29 and 30. In addition at different points on the line 28 an indication is given of the pressure of the gases contained in the cylinder at the moment the critical area of exhaust has opened. This indication of pressures is intended to show that as the engine speed increases the pressure of the gases remaining in the cylinder at the moment the critical area of exhaust is opened becomes greater and greater, as explained above.

It will be seen that the curve 30 is higher than the curve 29 at lower engine speeds and that thereafter it falls a little below the curve 29 and that from 1,000 R. P. M., to 4,800 R. P. M., the ordinates of this curve 30 only increase by 6°.

Consequently in such an engine, if inlet is timed to open at 18° after exhaust opens on the line 31, over the whole speed range of the engine, this timing of inlet opening will remain suitable for the introduction of the fresh charge into a completely void cylinder.

At the highest speed inlet opening is very close

to the moment when the void is left in the cylinder and at lower speeds there is never more than an interval of 6° between the occurrence of the void and the opening of inlet.

This reasoning is independent of the position of the exhaust orifice but such a timing will be necessary in the case of an engine in which both the inlet and exhaust ports are situated at the same end of the cylinder, for example, an engine having inlet and exhaust ports both operated by the same piston.

If the exhaust and inlet orifices are at the opposite ends of the cylinder, then inlet may be opened, strictly speaking, immediately the imaginary gaseous piston becomes separated from the end of the cylinder at which the inlet orifice is situated, so that it will be possible in such an engine to establish a fixed timing of inlet such that inlet opens immediately the critical area of exhaust orifice has been opened.

In the example considered this would be at 7° after exhaust opens. In practice, however, some latitude will have to be allowed and a suitable fixed timing of inlet opening could be established for such an engine at say 12°.

In the foregoing description only brief mention has been made of the period of delay that elapses before the burnt gases commence to leave the cylinder at their ballistic speed. It will be understood, however, that consideration of this delay is subordinated to that of the critical area of exhaust opening; but in order to make the description more clearly understood, the line 32 is included in Figure 3 to show the general situation of the delay that elapses before the burnt gases commence to move outwardly through the exhaust orifice or duct as a consequence of their mass exit from the cylinder. It will be understood that this mass exit commences at the end of the delay period in question, and that thereafter if the critical area of exhaust orifice is opened in a sufficiently short interval of time, the tail end of the issuing gaseous mass separates itself from the end of the cylinder with which it is in contact. It can also be imagined that at some very high speed of the engine, the critical area of exhaust will be opened in such a short interval of time that the burnt gases have not yet been able to commence their mass exit from the cylinder and such a condition is represented purely by way of example by the intersection of the line 32 with the critical area line in Figure 3 at 4,500 R. P. M.

In carrying out the present invention, it should be noted that the requirement is that the critical area of exhaust orifice shall be opened within the critical time interval, and this imposes the condition that the design of the engine must permit this result to be obtained and that the devices selected must be those which are capable of permitting the conditions to be fulfilled.

In applying the invention to any particular engine, this may involve radical alterations in design, including the design of the exhaust orifice controlling means and the exhaust orifices themselves.

For example, it may be necessary to provide a ring of exhaust ports all round the cylinder, and this will involve the use of two or more exhaust pipes since there must be no surfaces which oppose the exit of the burnt gases from the cylinder or which tend to reflect those gases back into the cylinder.

In general the requirements of the invention may be fulfilled irrespective of whether the stroke of the cylinder is relatively long or short, but since an increase in length of stroke relative to the bore will increase piston speeds, such a design will more easily permit the desired rapid openings of the exhaust orifices to be obtained.

Moreover, an increase in length of stroke will have the effect of increasing the critical time t.

In addition, the longer the stroke relative to the bore, the smaller will be the proportion of the cylinder volume of burnt gases that can be discharged from the cylinder before the critical area of exhaust has opened, and consequently the greater the ease with which the imaginary gaseous piston can be made to separate itself from the cylinder end at low engine speeds.

It should also be borne in mind that any exhaust ducts that are placed in continuation of the exhaust orifice will exert a controlling action on the issuing mass of burnt gases.

Further, it is desirable that after the critical area of exhaust orifice has been opened the necessary additional area of orifice should be opened to facilitate the escape of the burnt gases from the cylinder, since in this way the mean area of exhaust orifice is increased and the time interval occupied for the discharge of the imaginary gaseous piston will be reduced.

By means of the invention, conditions are established whereby an optimum torque and stability of the engine may be obtained over a desired speed range, and any suitable means described in the applicant's British or prior British specifications may be utilized in order to permit such a result to be obtained.

The invention has a particularly useful application to engines of the kind wherein the exhaust orifice closes later than the inlet orifice, such as engines in which both the inlet and exhaust orifices are situated at the same end of the cylinder and are controlled by the same piston.

By the use of the present invention, such an engine can be established in which as a consequence of the discharge of the burnt gases, a completely void cylinder is ensured.

Thereafter, by a suitable design of exhaust duct it will be possible to ensure that in the neighbourhood of the highest working speeds the return of the burnt gases is made to coincide substantially with the closure of inlet and in the interval between inlet closure and exhaust closure, and at all lower speeds the contents of the cylinder are protected from the return of the burnt gases which will then occur before inlet closes, whereby fresh gases which have passed through the cylinder into the exhaust system will be returned to the cylinder and may give a supercharge. Means capable of establishing such a protection against the return of the burnt gases have already been described by the applicant in prior specifications Serial Nos. 738,016 and 83,120, while in applications Nos. 38,826 and 46,805 the influence of the design of the exhaust duct upon the interval elapsing between the mass exit and the return of the burnt gases has been explained.

It has been stated above, that where numerical values are given in order to enable the present invention to be carried into effect, these values are practical values which are suitable for adoption. But in considering all these values, and the details of the curves shown in the figures it is again emphasized that these curves do not pretend to be accurate record of all the factors that can be considered to be involved, but are practi-

cal representations which can be used as a guide in carrying out the invention.

I claim:

A two-stroke internal combustion engine wherein the evacuation of the cylinder by the mass exit of the burnt gases is utilized for introducing a fresh charge, which comprises an exhaust orifice in the cylinder and means for opening and closing said orifice, the shape, dimensions and position on the cylinder of said orifice and the means for opening said orifice being so constructed and arranged that a sufficient area of said orifice is opened at such a rate at substantially the minimum working speed of the engine that the ballistic speed of exhaust is maintained for substantially the whole of the burnt gases throughout the evacuation of the cylinder at said minimum speed, whereby the above condition will be maintained at all higher working speeds.

M. KADENACY.

SAE *Quarterly* Transactions
April 1947 Vol. I No. 2
p. 204-216

The Pulsating Jet Engine

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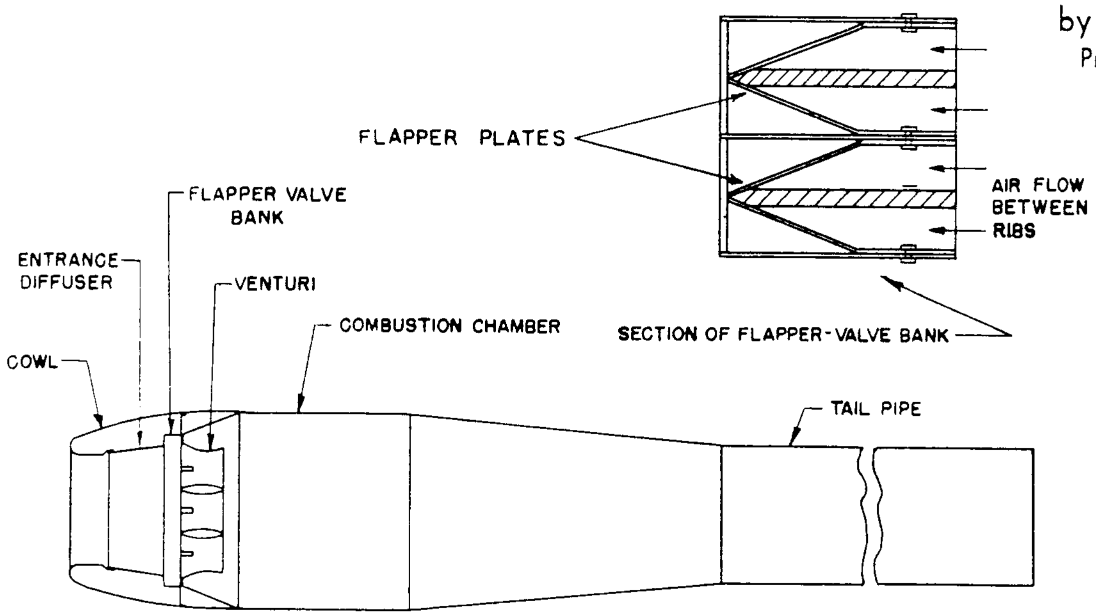


Fig. 1 – Schematic diagram of German V-1 engine. United States designation is PJ-31

THE pulsating jet engine has been given several names including "buzz engine," "resojet," "aeroresonator," and "pulse jet." It was first seriously scrutinized by American military and technical men after the initial firings of the German robot bomb, or V-1 weapon, in the late spring of 1944.

As a general type, the engine may be thought of as an intermittent, compressorless, aerodynamic powerplant, comprising few or none of the mechanical features of conventional stationary powerplants. In its simplest form, its operation depends only on the properties of atomspheric air, a fuel, a shaped tube, and some type of flow check valve, and not on the interposition of pistons, impellers, blades, or other mechanical parts whose geometry and motion are controllable.

The principles involved are closely related to the well known Kadenacy principle used in scavenging systems for 2-cycle reciprocating engines. In closer detail (Fig. 1), the engine is composed of a shaped metal tube fitted with one-way flow check valves at the forward end. The metal tube is shaped so that air flowing inward through the valves passes through a venturi, a cylindrical combustion chamber, a transition cone, and then a tail pipe to the exterior. The engine is started by introducing fuel and air into the combustion chamber, on the wall of which is mounted a spark plug for starting. The first explosion closes the air inlet valves and forces the tail pipe gases outward. Inertia effects in the tail gases cause an expansion below atmospheric pressure, and a fresh charge of air is brought in through the valves. The fuel, which is injected in a continuous spray, mixes with the fresh air and is ignited by the hot gases from the proceeding explosion. Operation proceeds without the benefit of ram air or operation of the spark plug. Pulsations give rise to the emission of a loud buzzing sound.

[This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Calif., Oct. 3, 1946.]

Early Evolution of Engine

History – Apparently, the earliest work along these lines was that done by Karavodine in 1908. He used a long, straight tube, one end of which was fitted with a flow check valve, the other end open and directed at the blades of a turbine. Inertia of the air column created low pressures and subsequent induction of fresh air after each explosion. Stodola records that this machine operated very inefficiently, delivering 2 hp with the consumption of 11 lb of benzene per hp-hr. This fact seems partially to account for its subsequent neglect.

Meanwhile, thinking had developed along the lines of using pure rockets for the propulsion of aircraft. It became obvious that it was impractical in many applications to use a reaction device which necessarily carried aboard the aircraft all the fluid which was accelerated backwards for propulsion. A scheme was devised whereby thermal energy was extracted from the rocket propellants and distributed ultimately not only in the propellant gases themselves, but also in a certain quan-

—Its Evolution and Future Prospects

tity of external atmospheric air. This air was to augment the mass of rocket gases expelled and thereby furnish a greater mass of "momentum medium." The proposal was to entrain and mix free air with the propellant gases after their expansion through a nozzle into an ejector-like device, after which expansion, the propellant gases still exhibited a high enthalpy. By mixing and further expansion, energy was to be transferred to the atmospheric air, causing its velocity to increase and a net forward thrust to result. Schemes of this type may be found in the patent literature on jet propulsion.[1]

[1] See *Aircraft Engineering*, Vol. 14, February, 1942, pp. 32-39. "Thermal Air Jet Propulsion," by Gohlke.

[2] See Main German Patent No. 523,655, April 25, 1931, "Method of Producing Reaction Forces on Aircraft," awarded to P. Schmidt.

FROM Karavodine's inefficient tube in 1908 through American developments after the recovery of German V-1 bombs, Mr. Edelman sketches the history of pulsating jet engines.

Work in the United States has centered on instrumentation, tube geometry, fuel injectors, air inlet valves, fuel properties, and performance estimation.

Realized performance is still low in comparison with predictions of best possible performance; but the author suggests use of the pulsating jet for helicopters having jets at the blade tips, for gliders, for starting turbines of aircraft gas turbine powerplants, and for auxiliary power with conventional aircraft.

L. B. EDELMAN is, at the present time, working on "Project Squid" at the **Palmer Physics Laboratory, Princeton University.** While serving for two years in the USNR, he was engaged in pulse-jet development work at the Naval Experiment Station at Annapolis, Md., and served for about six months as Bureau of Aeronautics' project officer in charge of this work. After receiving his BS in mechanical engineering from Louisiana State University, Mr. Edelman undertook graduate study on basic aerodynamics and thermodynamics at Harvard University. He received his MS in aeronautics from C.I.T. in 1943 while working as research engineer at the Jet Propulsion Research Laboratory there.

The Invention – Paul Schmidt, member of a consulting firm in Germany, reasoned that these augmentation schemes would bear no fruit, since ejector action followed the laws of plastic shock. His conclusion was that shock losses in mixing would cause a reduction in the gas velocity such that the effect of increased mass on momentum would be balanced, and no net thrust increase would be experienced.[2]

He then conceived the idea of transferring energy to the free air by means of combustion pressure applied to a plane interface between the "energy" and "momentum" (free air) mediums contained in a straight tube (Fig. 2). By such a piston-like transfer he claimed that no shock losses would result. Refilling of the tube was to be accomplished by means of the inertia of the departing air in the tube and through a set of check valves at the forward end. Similar ideas were expressed by Marconnet in 1909.

Work began in 1928. By April 25, 1931, Schmidt had obtained German patent No. 523,655 on this idea. Until 1935, the source of funds for part-time experimentation was consulting work, which he pursued with most of his energies. From 1935 forward, he proceeded with a small full-time group given only lukewarm support by the German air force, until in 1942 his version of the V-1 engine was turned over to Argus in Berlin for production. His own work continued until just prior to VE Day.

Origins in America – Information on German developments is hazy and incomplete; the history of American work may be related with slightly more assurance. It is not entirely clear, however, to what degree American workers, perhaps unknowingly, were influenced by the published literature on patents issued to Schmidt and others. It is clear, at any rate, that the earliest serious experimental work in America was started more than a decade after the issuance of Schmidt's patent.

E. B. Myer experimented in 1943 on what he called a "detonation engine," which was comprised of a simple metal cone. A small explosive charge was set off at the vertex, and the impulse was measured as a function of charge size and motor geometry. Apparently, his work was confined to single pulses, prior to recovery of the V-1 engines.

Late in 1943, a group at the Aerojet Engineering Corp. under the direction of Dr. F. Zwicky began consideration of a device called an "aeropulse." This was conceived of as a tube, one end of which was fitted with check valves; the entire tube was to be filled with a combustible mixture and burned under constant-volume conditions. Refilling was to be as in Schmidt's device, although there was a contrast in that only a small length

on the forward end of the Schmidt tube was to be filled with combustibles. Aerojet was given considerable financial support by the United States Navy's Bureau of Aeronautics, and work began early in 1944. To this group goes the credit for gaining the first wholehearted support of the armed services for work along these lines. At the same time, the Bureau of Aeronautics set to work its project at the United States Naval Engineering Experiment Station, Annapolis, Md.

By early spring of 1944 an engine based on the broad "aeropulse" conceptions was built and successfully tested at Annapolis by Lt. W. Schubert. This "valveless resojet," as it was called, had no moving parts, its check valve being a restricting tube whose operation was acoustical rather than mechanical (Figs. 3 and 4). Schubert's engine is believed to be the first pulsating jet built and run in this country.

A few months later, in the summer of 1944, the first V-1 engines were recovered, and experiments were initiated at a number of other government and private establishments.

Developments after Recovery of V-1 Engines

A considerable amount of active interest arose in this type of engine in view of its effective application in the V-1 weapon. Work undertaken in this country may be broken into several broad categories as follows:

1. Development and application of measuring techniques for study of engine operation.
2. Investigation of the effects of changes in tube geometry on performance.
3. Development of fuel introduction systems for better combustion.
4. Development of air inlet valves for longer life and higher air inlet capacity.
5. Testing fuels of widely different properties in the same engine in order to correlate fuel properties to combustion behavior in an effort to develop fuels producing higher combustion pressure.
6. Theoretical studies of engine operation.

Instrumentation – Measurement techniques for pulsating jets were necessarily the subject of development as well as application. Satisfactory techniques for measuring the time average of thrust, fuel flow, airflow, and body temperature were established after minor modifications of existing methods. Work was initiated to develop techniques for observing instantaneous effects, including wall static pressure and temperature, and flame and air velocities internally and near the tube entry and exit.

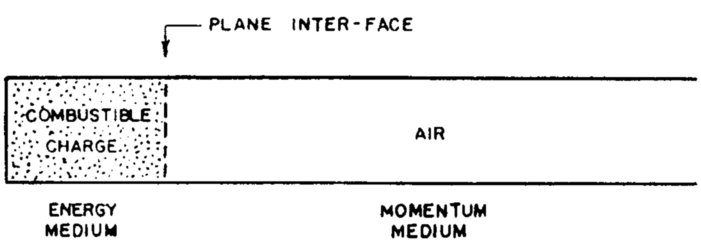


Fig. 2 – Schmidt proposal for augmentation by piston-like energy transfer to momentum medium

A detailed discussion of the work done on these instantaneous measurements is beyond the scope of this paper. Attempts were made to apply various mechanical, electrical, and optical measuring devices to this engine. In most cases, application of existing techniques required extending the useful ranges. For instance, pressure pickups in common use fell far short of the demands of this engine as concerned cooling for high heat transfer rates. A second example was in hot wire anemometry for instantaneous velocity measurements, existing equipment failed at lower velocities and temperatures than those encountered in pulsating jet operation under even static conditions. Broadening of the applicability of these instruments to include this engine was a slow and difficult process, and advancements were modest at best.

Engine Geometry – The effects of geometrical changes on performance were, on the other hand, very simple tasks experimentally, the difficulty being in establishing a rational theory to generalize the test results. By fixing the type of valve and fuel injection system, it was possible to select the major diameter or approximate design static thrust value as the fundamental independent variable, and to test the effects of the other dimensions. The following conclusions resulted:

(a) Below a certain "slenderness" ratio of total length to major diameter, operation could not be sustained.

(b) Optimum ratio increased as maximum diameter decreased; the range of values of the ratio lay from 5 to 12, in diameters from 6 to 22 in.

(c) Straight tubes or slight geometrical deviations therefrom were inferior for most types of valves tested.

(d) Venturis just downstream of the valve bank were useful chiefly in shielding the valve flapper plates and in creating turbulence beneficial to combustion.

Fuel Introduction – A limited amount of testing was done on varying the design and location of fuel injectors. In general, the information obtained was of value only when a given fuel under a fixed condition of preheat was specified. Data were obtained on a few types of fuel nozzles as follows:

(a) Fan spray in a plane perpendicular to the engine longitudinal axis.

(b) Linear and swirling cone sprays along axes parallel to the engine axis.

(c) Discrete linear jets oblique to the engine axis.

It was found, as might be expected, that sectionwise distribution of the fuel jets in any given plane was critical for type (b), important for (c), and somewhat indifferent for (a). Limitations on the distance of the nozzle location downstream of the valve bank were not carefully studied, but

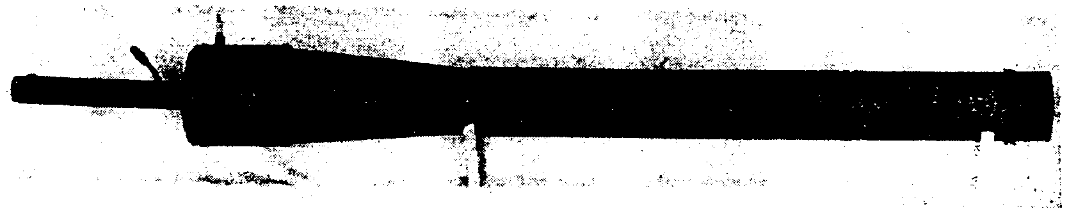


Fig. 3 – Side view of Schubert's pulsating jet engine utilizing acoustical air valve composed of tuned restricting pipe

effects on performance were small in the range from zero to about one-half the engine major diameter. Location of the nozzles from about one-half to one engine diameter upstream was satisfactory for fuels exhibiting low volatility and long ignition delay.

Almost none of the questions of optimum fuel introduction design were definitely answered. Designs for higher nozzle pressure at the same fuel flow consistently yielded better performance due to atomization effects. Uniformity of distribution was also important. In spite of the limited amount of testing to date, the work done has had gratifying results primarily in the reduction of fuel consumption and secondarily in thrust improvement.

Air Valves – Of the several work categories listed, that of valve design yielded effects which were larger and more significant than those of any other. Development work was directed at two principal objectives as follows:

(a) To increase the effective open area of the valve bank in order to reduce pressure losses and to increase air capacity.

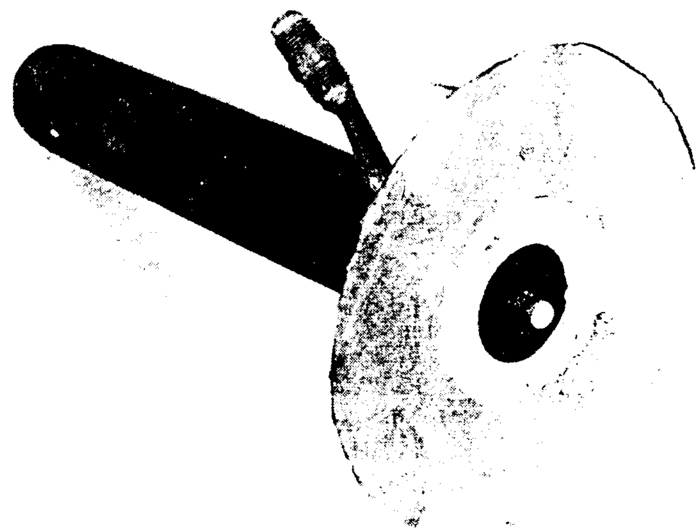
(b) To increase the effective valve life. (Thrust and specific fuel consumption deteriorated rapidly after about 30 min operation of the V-1 engine, due to damage in the valve flapper plates.)

In the recovery of the first intact V-1 engines there was provided a definite starting point for mechanical valve developments in this country. The inefficient aerodynamic characteristics (Figs. 5 and 6) and the short effective operating life were outstanding defects in this design.

Work on the effect of seating adjustments of the valve plates in the original German design suggested ideas leading to the later work. By adjustments of the flapper plates, it was possible to provide different values of the seating pressure of the plates on the grill, or to provide a gap between the two. As the seating pressure was reduced and the plates began to seat lightly or have a gap up to about 1/4 in., the peak static thrust increased continually. Airflow measurements showed that air capacity increased in this direction, maintaining approximately constant fuel-air ratios and specific fuel consumption. With the thrust increase was found an increase in the fuel-flow range from idle to flood-out conditions. This confirmed beliefs that the tube of the V-1 engine was not being furnished all the air it could draw in and handle, due to poor design of the valves.

In the same tests, it was noted that lightly seated plates gave better life than those with open plates. Considering the valve as a driven vibrating

Fig. 4 – Closeup view of Schubert's propane injector and tuned pipe



system, it was clear that the seating velocity would be greatest when the seating point coincided with the system's neutral point. If valve failures were due, then, entirely to fatigue from repeated impact, the open-plate condition would have had lower seating velocities, lighter impact loads in each cycle, and consequently longer life. This was contradicted in the tests, and it was concluded that failures were due to impact fatigue and also flame leakage upstream through the seats.

From the indication of air-capacity improvement came a series of valve redesigns. This work was intended to increase the open area associated with each flapper plate and its backing grill, and to increase the number of flapper plates feeding air to a given cross-sectional area. To achieve the latter effect, the plane grill was "dished" into various concave and convex geometries, including shallow and deep rectangular boxes, truncated cones, and truncated cones concentrated within one another. The thrust increases due solely to valve redesign can be safely set at a minimum of 50 to 75% improvement in peak static thrust, and were obtained by increasing the effective open area through the valve system from about 20 to 50% of the cross-section. Again, turbulence played a prime role here. In some valve designs, the open area was as high as 80% with no improvement in thrust, apparently due to a lack of the proper turbulent conditions. Little or no use was made of the existing knowledge of turbulence in dealing with this problem. Consequently, improvements have been considered significant, but well below those to be reasonably expected as possible.

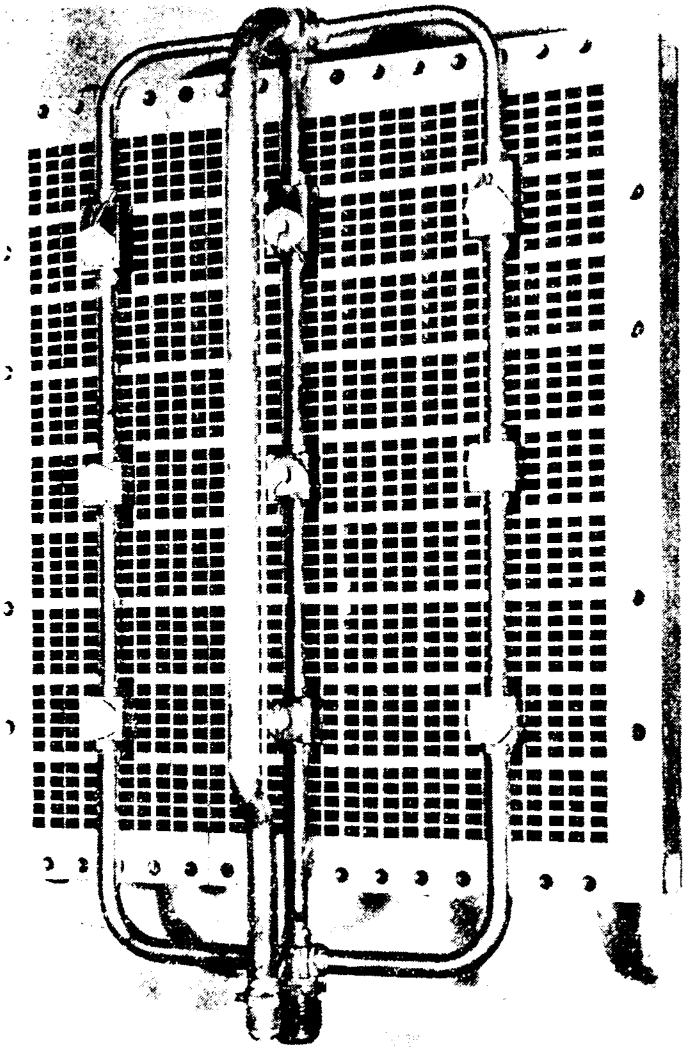


Fig. 5 – Upstream face of PJ-31 valve bank showing tubes for fuel and starting air injection

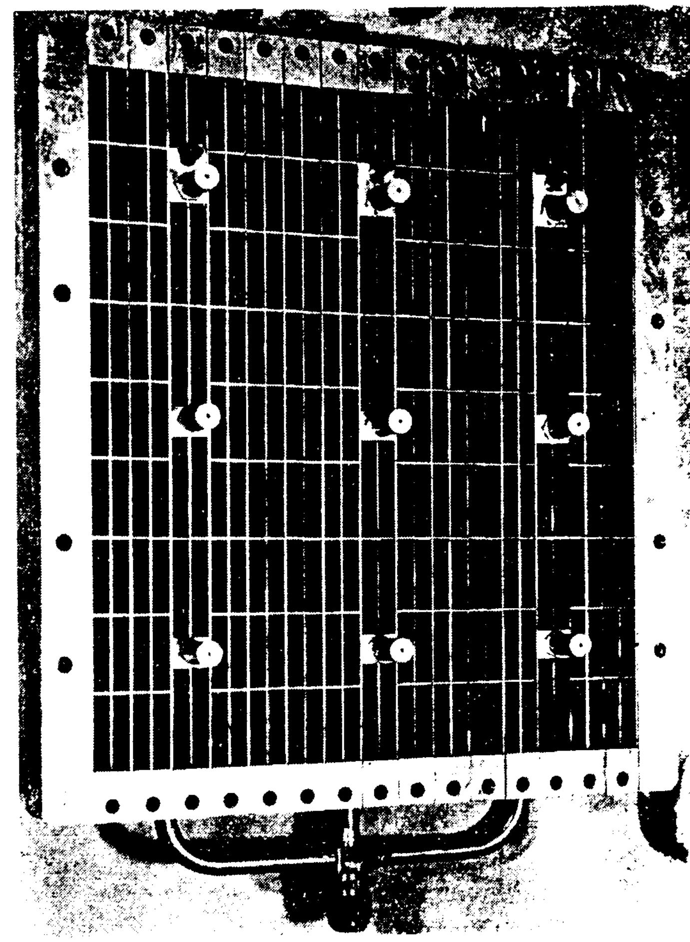


Fig. 6 – Downstream face of PJ-31 valve bank showing closed flapper plates, fuel nozzles, and air nozzles (adjacent to upper fuel nozzles)

Concerning valve life, recognition of the causes of failure led to effective steps along lines to increase the useful operating period as follows:

(a) By providing a soft grill seat to reduce impact of the plates in closing and to inhibit flame leakage forward. (This was accomplished by coating the seats with neoprene about 0.015 in. thick.) This work was done at the Cleveland Laboratory of the NACA.

(b) By separating the effects of heat and impact by a sandwich design of valve in which the inner plate takes most of the impact and the outer plate most of the heat. (See Fig. 7.)

(c) By geometrical design such that valve plates operate cold, combustion pressures being applied through small cushions of cold air.

(d) By geometrical design such that the amplitude of valve-plate motion and, consequently, seating velocities were reduced.

It is noteworthy that metallurgical methods were not prominent in this work, perhaps through neglect. Most of the successful work was accomplished with the use of stainless and spring steels.

In contrast to the starting point of about 30 min. effective valve operating life was extended to several hours. Designs for life from 6 to 10 hr were realized. Workers in this field enthusiastically viewed the possibility of putting this problem on the "25-hr check" basis.

In passing, it is interesting to mention valve constructions not incorporating the metal-spring principle. Reference to the valveless engine of Schubert has been previously made. Another successful construction was that using a simple rotating butterfly tuned to the engine frequency. (See Fig. 8.) Both engines exhibited very long useful operating periods. The latter engine exhibited good thrust and specific fuel consumption but has required a synchronous electric motor as a drive.

Fuel Studies – A series of preliminary tests was performed to study the effects on performance of various fuels with widely differing properties. Contemporary engines exhibited peak combustion pressures of the order of about 2 to 2.5 atmospheres abs, as compared to ideal constant-volume values of about 9 to 10 atmospheres at corresponding precompression and fuel-air ratios. It was reasoned that, for a fuel exhibiting a given overall rapidity of heat release, the peak pressure obtained would be determined by the effective inertia of the column of gas. The degree of the gas column's tendency to remain at rest as combustion pressure was applied to its boundary common with the combustibles, was seen to control the degree to which the volume of combustibles increased as

heat was released and combustion pressure rose. Conversely, with given gas column inertia restraint, the peak pressures were expected to increase with rapidity of heat release.

With these general ideas as a guide, a considerable number of fuels and fuel additives was tested and compared to 62-octane fuel chosen as a norm. Reciprocating engine fuels were tested in a range from 100% cetane to 120 octane. Kerosene, benzene, methyl and ethyl ether, propane, and a variety of related compounds were tried. One worker reported tests on addition of nitroglycerine to gasoline. Generally, the program was designed to probe out expected large effects and determine the fuel properties causing them. In this sense, the program was negative; improvements were barely discernible in the shadow of experimental errors, which were held within a very few per cent. A few significant facts discovered are as follows:

(a) The pulsating jet engine of contemporary design ran on almost any common fuel with negligible variations in performance.

(b) Physical and chemical properties of fuels affecting operation of diesel and otto cycle engines had no marked effect on the pulsating jet engine. Principal differences were in the degree of body heating and in the rapidity of valve destruction.

(c) Nitropropane gave about 20% improvement in thrust with about a 100% increase in specific fuel consumption. Favorable comparisons at altitude, expected on the basis of the oxygen content in the fuel, were not realized.

(d) Changing from liquid to gaseous fuels was possible by minor changes in fuel injectors.

Theory – Theoretical studies of pulsating jet operation were begun in the spring and summer of 1944, and by late 1944 reports from the National Advisory Committee for Aeronautics and from some private companies had appeared. In all these reports, the assumption was made that the entire tube was filled with combustibles and burned under constant-volume conditions. These studies were made with an almost complete lack of supporting experimental data; unfortunately, the assumptions made were in violent disagreement with operating conditions, and the calculated performance also showed disagreement to a similar degree. This early work was valuable, however, in laying a foundation for later work.

A subsequent analysis by N. P. Bailey and H. A. Wilson was benefited by the availability of limited test data. These studies made quantitative estimates of the effects of heat release rates as discussed in connection with fuels. Further analyses, and empirical data as they slowly became available, made possible the rather advanced calculations by a group at New York University under the direction of Prof. R. Courant.

The earlier of their reports[3] concerned itself with the application of modern gas dynamics to the prediction of impulses resulting from detonating and burning the combustible mixture in several different ways. The studies made were valuable not only in casting serious doubts on previously unquestioned ideas of detonation effectiveness in this engine, but also in leading directly to the formulation of new and more important optimum problems in combustion. These will be discussed in a later part of this paper.

The latter of these reports[4] was an attempt to provide an accurate and yet tractable mathematical treatment including the effects of all important engine phenomena then known. It is yet too early to judge the importance of this contribution because of its recent issue and because its value is in direct proportion to the availability of fundamental empirical data.

A comment is in order at this point to indicate the effect of the accompanying difficulties on the development of pulsating jet theory. By comparison to steady-flow devices, these difficulties were very great. One was the fact that nonlinear oscillation problems were involved. A second difficulty lay in the inability of the existing physical understanding of gas dynamics to predict for many empirically unfamiliar situations the occurrence of shock and detonation waves. These were only

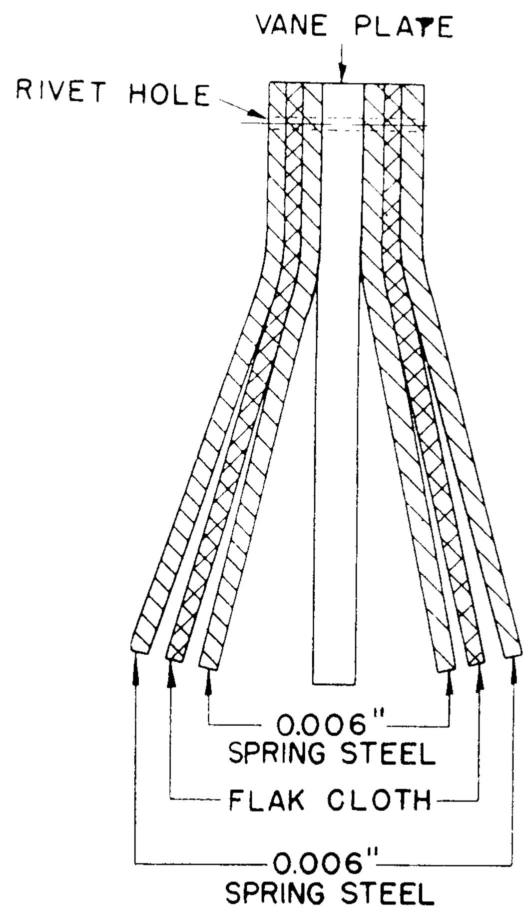


Fig. 7 – Sketch of sandwich-type arrangement of flapper plates

[3] See "On Effectiveness of Various Modes of Detonation or Combustion." Published by Institute for Mathematics and Mechanics, New York University, New York City, February, 1946.

[4] See "Gas Dynamical Formulation for Waves and Combustion in Pulse Jets." Published by Institute for Mathematics and Mechanics, New York University, New York City, June, 1946.

two of several real and ever-present obstacles along the path to a theory of engineering value. These difficulties, and their consequent delay of the development of the theory, have placed this work on a slow, empirical basis as compared to the steady-flow engines, where general theory and correlated detached experimentation have greatly accelerated progress.

Present Status

Knowledge of Present Phenomena – Certainly, with any such engine whose realized performance is low relative to its possibilities, detailed knowledge of the physical mechanisms of its operation comprises an important part of the present state of the art.

Characteristic of this engine type is the fact of pulsation. The frequency spectrum of emitted sound from a small experimental engine (Fig. 9) shows that the fundamental and both odd and even harmonics are prominent. The even harmonics may be expected because the valves are open during a part of the cycle, and this end of the tube acts as a velocity loop and as a node. With the choice of a reasonable average temperature, the fundamental frequency is closely approximated by formulas for either the quarter-wave organ pipe or the Helmholtz resonator – use of the latter requiring a rather unnatural choice of capacitance and inertance. Clearly, simple acoustical resonance is only a crude description of the operation of this engine.

Valve-motion studies of the same experimental engine (Fig. 10) emphasize the relatively small open area of the valve bank as compared to its total area. Furthermore, the valves are closed during about one-half of each cycle (Fig. 11), profoundly affecting the aerodynamic design of the inlet diffuser.

A plot of wall static pressure against engine length at various ram pressures (Fig. 12) throws more light on the pulsations and on the valve aerodynamics. The average static pressures are positive at the forward end, indicating that the corresponding instantaneous pressure loop must swing farther above atmospheric than below; this condition cannot, of course, be deduced by ordinary acoustical reasoning. Further, the average pressures at this end are rising much more slowly than the ram, indicating unduly high losses in ram precompression across the valve bank, and possibly also poor combustion at higher rams.

In spite of the measurement difficulties, it is possible to plot (Fig. 13) the pressure-time diagram of the combustibles with some degree of accuracy. High velocities, instantaneously of the order of 1500 fps, of the tailpipe gases cause the combustibles to expand about 5 psi below atmospheric pressure. This causes simultaneously the induction of fresh air through the valves and also a complete reversal and forward flow of the tailpipe gases. This forward flow is at least partially responsible for the small precompression of about 2 psi preceding combustion of the new charge as indicated by the plot of single-explosion pressure in Fig. 13 under static conditions.

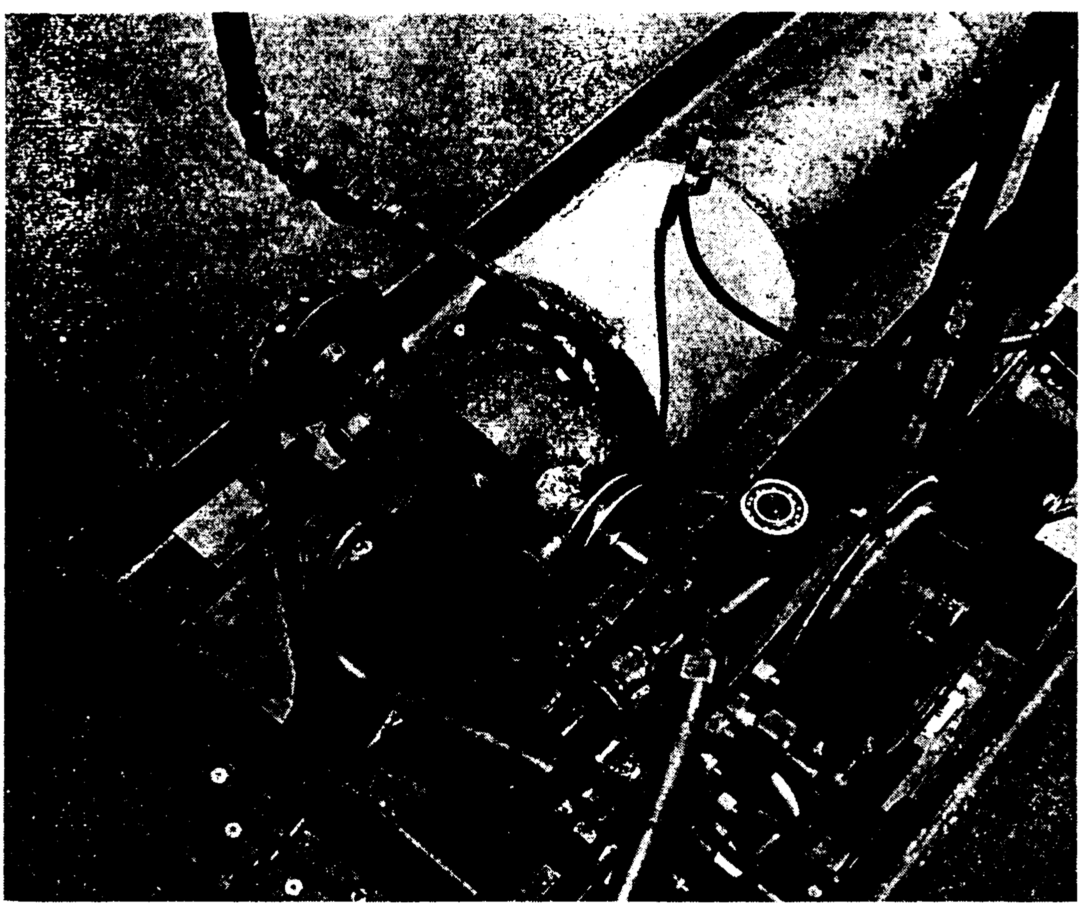


Fig. 8 – Rotary butterfly-valve engine with synchronous motor drive

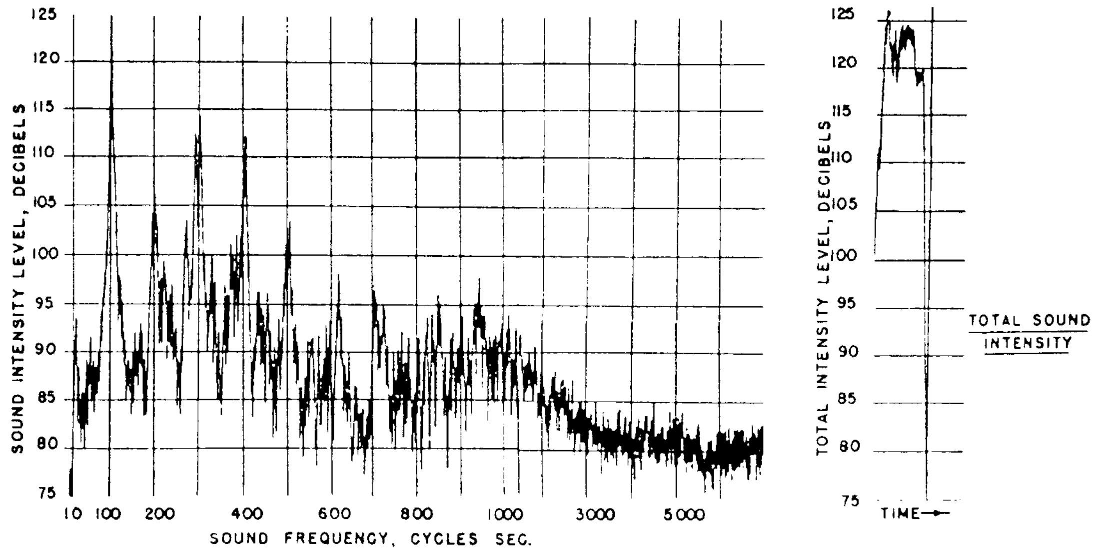


Fig. 9 – Frequency analysis and total intensity of sound emitted by 6-in.-diameter engine operating at peak thrust

The back-flow condition is quite marked. A test was made in which a tracing powder was sifted into the jet of a running engine at the exit plane. On studying the tube after shutdown, it was found that the tracing powder was deposited on the tube walls for a distance of two to three diameters upstream of the injection plane. Apparently, then, at low forward speeds approximately as much fresh air enters through the tail exit during each cycle as through the valves. This mass of air augments the mass of air accelerated and increases the tailpipe inertia effects partially responsible for the precompression and combustion pressure buildup. If not properly controlled, its effects on thrust at increasing forward speed will be detrimental. Furthermore, data taken at high rams in which air is ducted only to the valves and not to the engine's exterior will be optimistically incorrect since the backflow will be more favorably fed with static air. At speeds of 300 to 400 mph at sea level, the effect on gross thrust may be estimated at about 20 to 40%.

Combustion phenomena in this engine are but poorly known and understood. Flame studies were made as one of the optical observations mentioned previously by the use of high-speed motion picture cameras photographing perforated metal and glass-walled engines. Some of these observations are shown in Fig. 14 and seem to indicate that the fresh charge is ignited in the mid-region of the engine combustion chamber. That the charge is in condition to burn seems due to the fact that the fresh air is drawn through the valves into the fuel spray just behind, and subsequently flows along the straight section toward the cone, during which period there is time enough for atomization and mixing to occur. As the downstream boundary of the combustibles travels into the cone, the hot tailpipe gases have been caused to reverse their direction and flow upstream into intimate contact with the fresh charge. Ignition seems to occur at the boundary and propagate forward and inward from the walls where residual hot gases collect in the dead-water regions behind the venturis. Clearly, the flame propagation is three dimensional; however, the one dimensional plot of Fig. 14 is probably valid for the nonburning gases. The important feature of this plot is that the boundary between burning and nonburning gases has been sharply accelerated aft before burning is complete, showing the inertia effects of the tail gases on combustion pressure buildup.

The influence on test stand performance of all these effects is shown in Fig. 15. These are even more vivid in the thrust variation with Mach number in Fig. 16. This is a plot of early German test results on the V-1 on which little or no work was done to correct at high speeds the inefficient aerodynamic character of the early air-inlet valves of the diffuser designed for steady flow, or to overcome the loss of tailpipe augmentation. The thrust variation with altitude (Fig. 17) at constant indicated air speed also shows these effects.

Best Performance – Comparison of present performance levels to early levels shows the results of the work done to date. Table 1 indicates the best known figures for the separate items of static performance.

It should be noted carefully that these values do not necessarily correspond with those of any single engine, but are the best figures known for the separate items. Almost without exception, however, engines exhibiting the best or nearly the best of the single items have also shown in the other items substantial improvements which closely approximate the figures quoted.

Future Prospects

Problems and Trends – Considerable emphasis has already been laid on problems, the partial solution to which has been responsible for the considerably greater performance cited above. Future work should go forward in clear view of these problems, of their perspective as well as of their

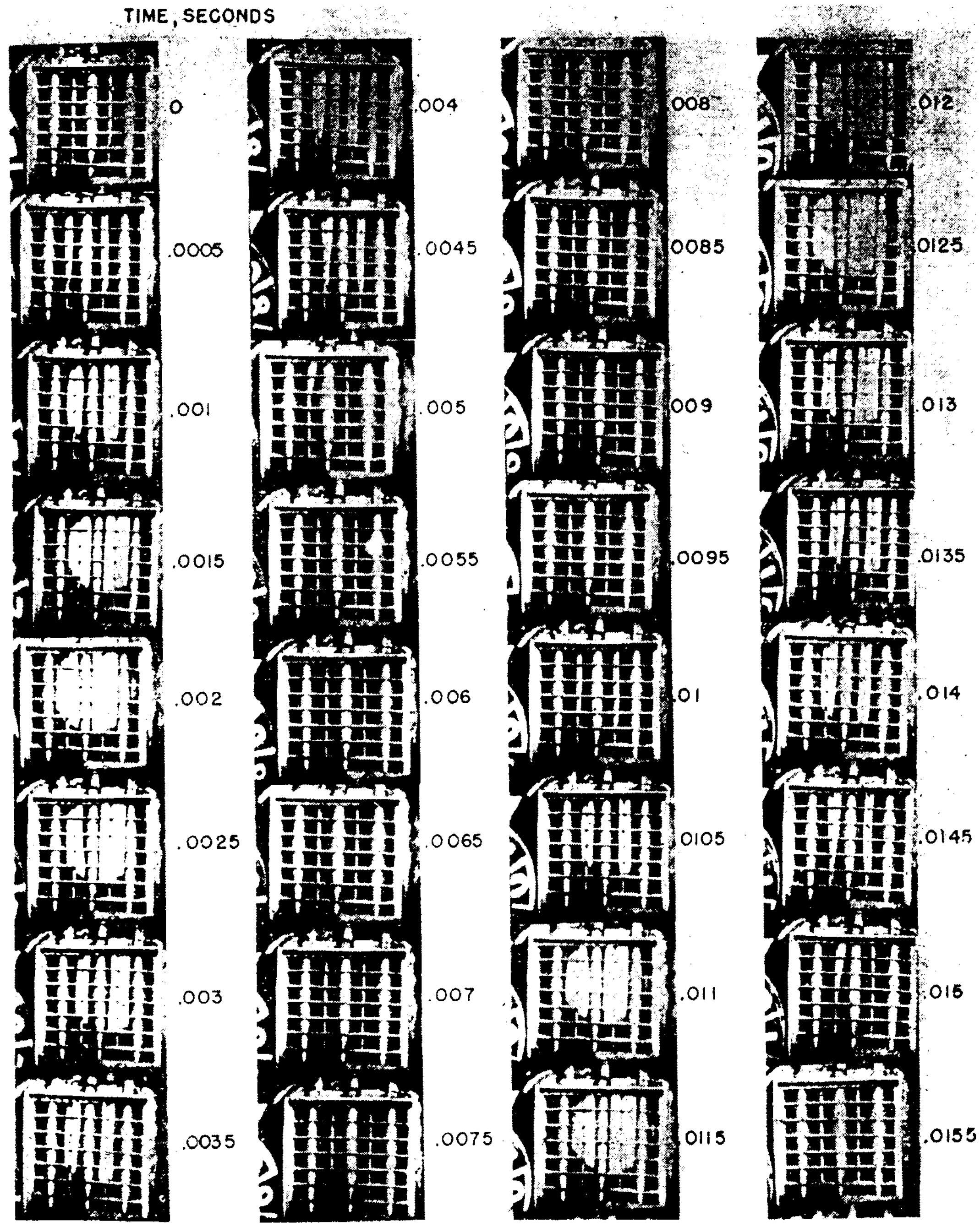


Fig. 10 – High-speed motion pictures of flapper-plate motion graphed in Fig. 11

details. In the order to which a degree of progress has been made in their solution, the more important may be listed as follows:

1. Air-intake valve design.
2. Combustion.
3. Tailpipe phenomena.

Valve Design – Progress due to valve redesigns has been marked, as noted previously. Further work along the same lines is indicated since the most effective present designs allow an open passage of only about 50% of the projected area. Furthermore, each of the many passages through a valve bank presents to the incoming flow unfaired blunt metal edges, and to the departing flow an abrupt change in cross section whose boundary is the sharp edge of the valve flapper plate. The argument is advanced that the turbulence so produced is essential for combustion. However, the cost in pressure losses for this turbulence is exorbitantly high, as can be seen in the poor pressure recovery with increasing ram pressures. (See Fig. 12.) Consequently, new designs are needed which admit larger quantities of air and simultaneously impart, with minimum pressure loss, the proper scale and intensity of turbulence for optimum flame propagation. It would seem that this duality of purpose of the valve bank would lead to inefficiency in each action; consequently, it is suggested that the valves themselves be designed for the former purpose only and in such a way that their open area is 80% or more of the section. Then other means should be provided to create the necessary turbulence with smallest possible losses.

Combustion – The combustion problem remains almost untouched experimentally except for fuel

introduction measures intended to eliminate gross wastes of fuel due to lack of participation in combustion. In fact, formulation of the problem is not yet definitely outlined.

Consider a slender straight tube closed on its forward end. A small fraction of the length, measuring aft from the closed end, is occupied by a combustible mixture of fuel and air (Fig. 2); the remainder of the length is occupied with air and inert products of combustion. It is desired that thermal energy from burning the combustible mixture be transferred to the entire gas column and be manifest ultimately in the form of kinetic energy of translation. Moreover, it is desired that the processes of extraction of the thermal energy and of its transformation to mechanical energy proceed with the least possible losses. Obviously, both thermodynamic and aerodynamic considerations are necessary as regards addition and extraction respectively, since the "piston" in this engine is a column of air. For example, from thermal considerations alone, the heat should be added to the combustibles under constant volume conditions; this would result in a momentary condition of a pressure differential of several atmospheres across the interface between the burning and inert gas columns. As an aerodynamic consequence, the extraction and transformation process would go forward irreversibly and with losses due to severe shock waves emanating from the plane of the pressure discontinuity. For the simple case considered, optimum conditions of heat addition and extraction are mutually self-exclusive. Suppose that heat is added so that the volume of combustibles is in-

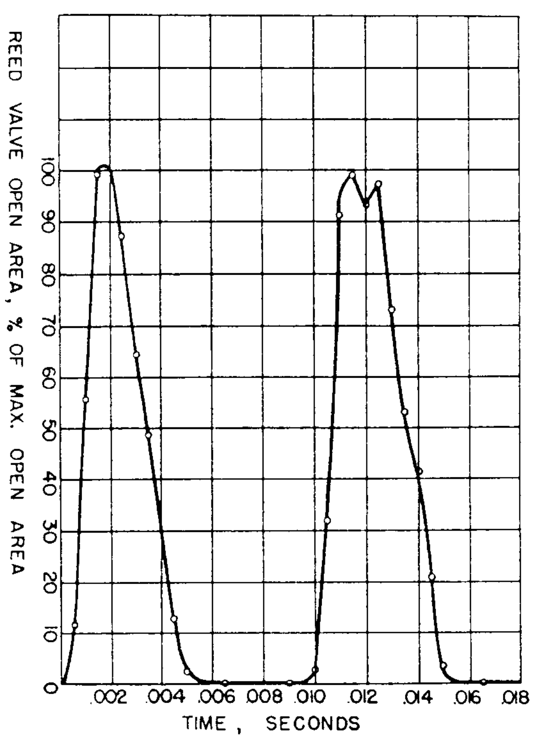


Fig. 11 – Plot of open valve area against time for typical small experimental engine, showing that valves are closed during about half of each cycle

Table 1 – V-1 Engine

	1944 and Prior	July, 1946
Thrust per Unit Major Cross-Sectional Area, psf	250	570
Specific Fuel Consumption, lb per hr-lb	4.0	2.6
Effective Valve Life, hr	0.75	7.0 and up

creasing during heat addition; then the aerodynamic extraction is proceeding more, and the thermal addition less effectively. Many modes of this process may be formulated; a limited analytical treatment of a few of these modes has been mentioned.[3] Unquestionably, a great amount of carefully coordinated theoretical and experimental work remains to be done before even a partial solution to this problem is achieved, before the possible advantages of this engine's cycle over the cycle of the steady-flow engines begin to be exploited.

A number of development trends have been proposed; these come forward from the conviction that presently realized pressure ratios of about two or three atmospheres can be raised in practice to six or seven of the ten atmospheres ideally possible, and with an improved efficiency largely overshadowing the shock effects mentioned. Among these proposals was that of using fuels with proknock and similar characteristics, as previously mentioned. Others are:

(a) Use of artificial ignition timed by the engine pressures, causing ignition at one or more points in the charge so as to achieve faster overall heat release.

(b) Use of traveling flame and detonation fronts to achieve precompression in the unburned portion of the charge.

(c) Use of timed intermittent fuel injection, as opposed to the present continuous injection, to eliminate rich, slow-burning regions of the charge.

(d) Control and promotion of turbulence to achieve higher apparent flame speeds and greater release rates.

(e) Use of faster-closing valves to eliminate combustion pressure losses upstream through the valve bank.

All such developments could be more realistically evaluated and pursued, however, if the present "natural" mode of ignition and flame spread were well known and understood.

Tailpipe Effects – Reference has been made to the effects of the gas column inertia in the tailpipe, namely, in causing a slight precompression of the combustibles, in causing combustion pressure rise, and in causing the induction in each cycle of a fresh charge of air into the tail. In addition to the inertia effects, the distributed elasticity or softness of the gas column is also significant. The oscillatory motion of this column is inherent in the pulsating jet and not normally encountered in the steady-flow engines; consequently, the fullest use of the effect should be made.

The problem itself may be stated simply: It is

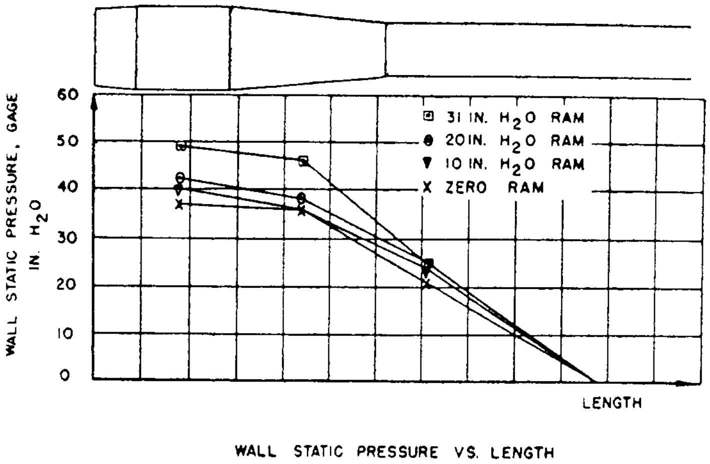


Fig. 12 – Wall static pressure versus engine length of PJ-31 at various ram pressures

desired to intensify all the known effects of the tail gases, since they are beneficial. A qualification need be made only as regards the method of introduction of fresh air to the tailpipe. The present method, that of drawing air from the free stream surrounding the tailpipe, is not satisfactory, especially as forward speed is increased, since the cost is high in energy and momentum to reverse the free stream and cause it to flow forward in the tail.

In order to accomplish these objectives, it is suggested that the tailpipe be fed fresh air through suitable ports in the engine tube from some point on the engine or aircraft where the flow is undesirably stagnant, such as the following:

(a) At a point where boundary layer buildup is excessive.

(b) Through ducts designed to relieve stagnation pressure on the valve bank when the flappers are closed.

(c) At points on the intake duct lips where stagnation is undesirable.

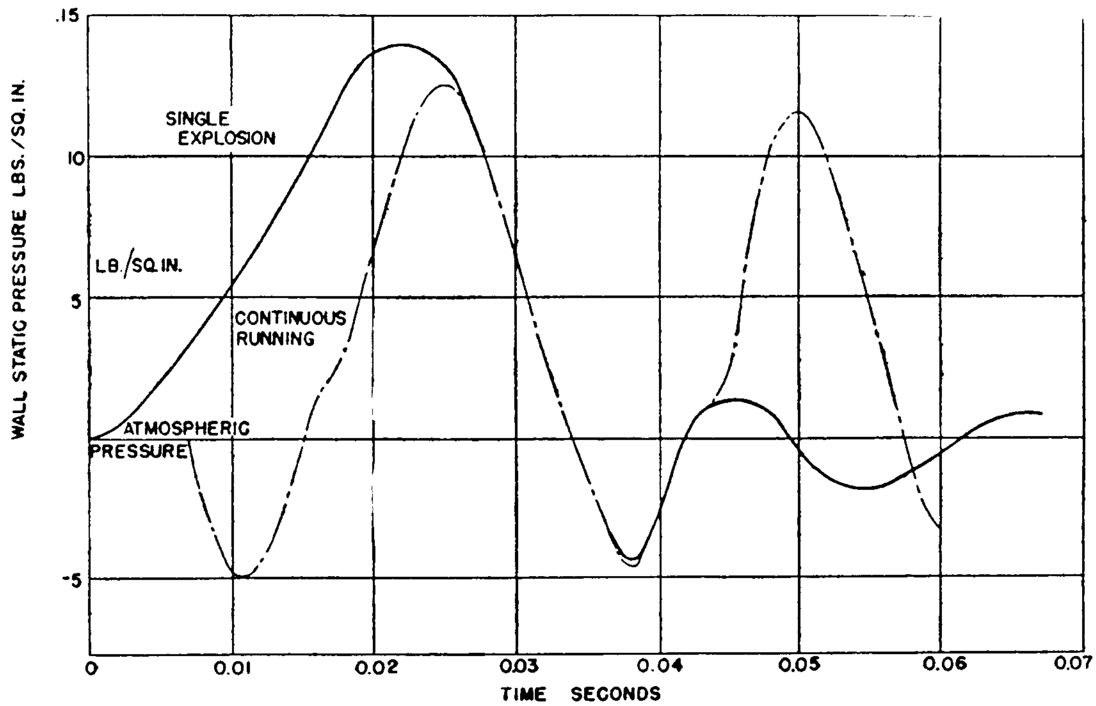


Fig. 13 – Plot of cyclic combustion pressures against time with superimposed single explosion in PJ-31 engine

Performance Possibilities – In most types of heat engines, including complete propulsive systems, it is possible to calculate theoretical efficiencies and outputs by means of idealized thermodynamics and fluid mechanics; realistic appraisals of losses may be made by resort to other analyses and to empirical data. In this way, the development engineer is able to make a good estimate of the upper performance limit of his engine, and by reference to the realized performance, he is then able to adjust intelligently the direction and intensity of his efforts. As previously mentioned, the difficulties in making these analyses for the pulsating jet engine are very great. While excellent work has been done along these lines, the

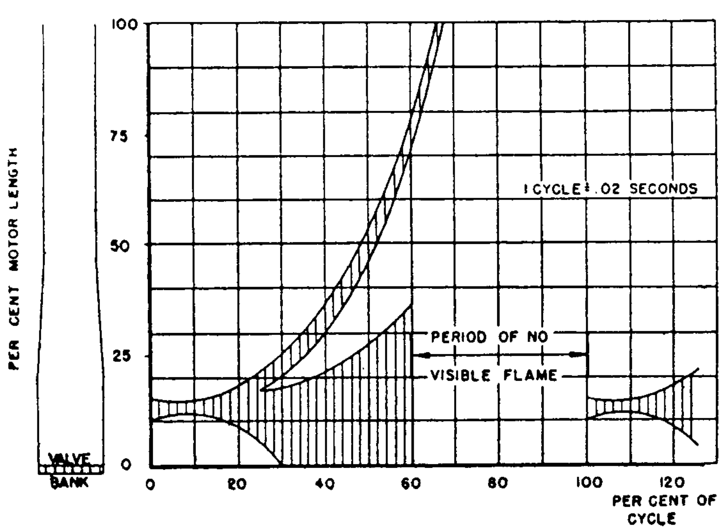


Fig. 14 – Flame motion versus time in PJ-31 engine

available estimates are much less reliable than those for the steady-flow propulsive systems such as the turbo-jet.

However, by careful screening and study of the available theoretical and empirical data, and with the inevitable resort to intuition concerning the degree of solution possible for the several development problems, an estimate of possible static performance, Table 2, can be made with some degree of assurance.

The purely speculative estimate above is based on an anticipation that present work on tailpipe effects will be successful, that the pressure ratios measured in single-pulse combustion in tubes can be realized in continuous operation, and that the air-capacity improvements indicated by late designs will be extended to the ultimate. The figure for fuel consumption follows after a number of conservative assumptions are made regarding the character and magnitude of thermal energy losses.

Applications – A number of uses for this engine suggest themselves because of its outstanding inherent advantages in design and performance:

1. The mechanical design is by far the simplest and least expensive known for operation statically and at forward speeds up to well in excess of half sound speed. Present and contemplated designs do not require specialized high temperature alloys or expensive machining operations.
2. Starting, throttling, and shutdown are dependable and easy to effect; a simple ignition system is required only for starting; control is dependent only on fuel flow and not on external governors or other auxiliaries.
3. Any common liquid fuel may be used without redesign or change in parts and without noticeable performance variation.
4. Dry weight is favorable in comparison to all air-consuming engines throughout a wide speed and altitude range.

Pertinent to this discussion are the undesirable features of present designs, as follows:

1. The fuel consumption is relatively high, the best figure to date being about 2.6 lb per hr-lb under static conditions.
2. Noise and vibration in operation are unpleasant though not generally dangerous or destructive.
3. Operating life of the valve elements is short.

These features are, however, subject to elimination or great reduction in degree of undesirability. Consequently, applications may be contemplated on the basis of present designs or on those to be reasonably anticipated in the near future.

As a principal source of propulsion, the pulsating jet is outstandingly suited to most types of pilotless aircraft, including those useful for military application, for transport of mail, and for flight research. As a sidelight, the avocational pursuit of model aircraft construction may be mentioned.

Studies of the design of helicopters with jets at the blade tips have been encouraging. The pulsating jet is especially interesting because of its high performance in a wide range of speeds, including static conditions, and because of its independence of the need for a mechanical compressor at the hub. A suggestion has been made that multiple short units be placed at the tips and be fed with air scooped in either at the tip leading edge or at the hub and carried through hollow blades to

Table 2 – Best Realized and Estimated Possible Performance

	Best Realized Performance	Possible Performance
Overall Air Fuel Ratio	25 to 35:1	50 to 60:1
Pressure Ratio	2 to 2.5	4 to 5
Thrust Per Major Cross-Sectional Area, psf	570	800 to 900
Specific Fuel Consumption, lb per hr-lb	2.6	1.5 to 1.8

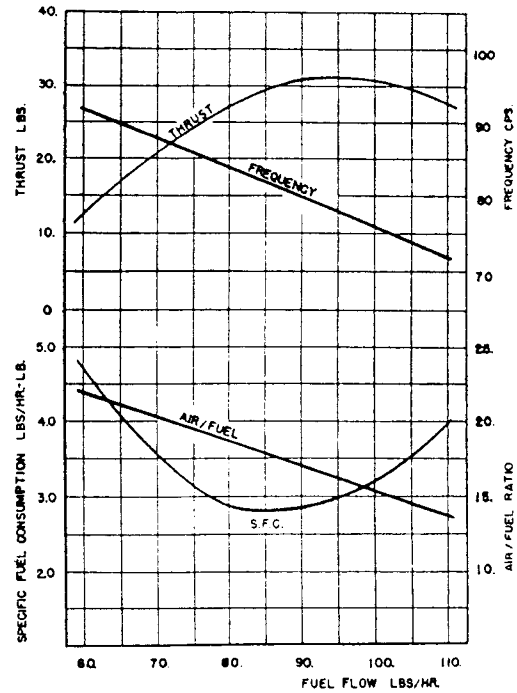


Fig. 15 – Performance plot for typical small experimental engine. Air-fuel ratios are based on air admitted through valves only and not at tail exit

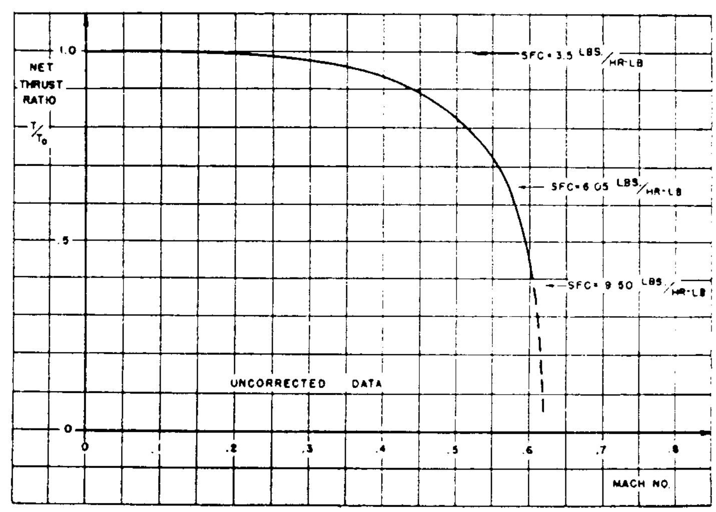


Fig. 16 – Early German aerodynamic data showing detrimental effects on net thrust of poor aerodynamic design of completed V-1 engine

the tips. Several mechanical designs are contemplated, and these give promise that this application will be even more attractive than those for conventional aircraft.

A helicopter powered in this way has several advantages over the designs using reciprocating engines. Elimination of the heavy engine and its gearing to the blades would result in great savings in dry powerplant weight, in space, in first cost, and in maintenance cost. The absence of the large torque transfer from the rotors to the fuselage would reduce structural weight and possibly obviate the necessity for the tail rotor. While fuel consumption would be higher on the basis of power delivered, the power required for the same performance in speed, climb, and payload would be much reduced. Furthermore, since use could be made of the least expensive fuels known, the operating cost for equivalent performance might be quite favorable. Undoubtedly, the promise for this application is sufficient to warrant serious effort in design studies and experimental constructions.

This engine is also well qualified for application to gliders in order to eliminate the present launching problems and to lend flexibility to flight and landing procedures. Aerodynamic inefficiency in the soaring condition due to the engine's presence could be reduced to a negligible amount by simple aerodynamic and mechanical design.

As an auxiliary powerplant for conventional aircraft, the pulsating jet may render service in a variety of ways. Consider the typical single-engine light aircraft in which engine failure is one of the primary hazards at least in the mental sense. In such aircraft, the pulsating jet could be installed, perhaps as an integral part of the load-carrying fuselage structure, so that it could be quickly started in an emergency. A relatively small and light engine could provide thrust enough for speeds of about 15 mph above stall speed for considerable periods. Noise levels of present designs would probably restrict usefulness to emergency conditions, although when effective muffling is achieved, the applicability could be broadened to include power boosts for take-off from short or muddy fields or for over-loaded conditions. The possibility also exists that, with simple mechanical design, the thrust could be reversed in direction to effect speed reduction for landing in small fields.

In passing, it is interesting to speculate on applications other than the propulsion of aircraft. The Swiss have developed a blowerless heating unit which operates on pulsating jet principles. The prospect exists for the incorporation of one or more small pulse tubes in aircraft gas turbines for starting by directing the jets into the turbine blades. Others may be mentioned which go afield even further from our subject.

Conclusion

An attempt has been made to outline with perspective the basic concepts of the pulsating jet engine, the work done to date, and the problems of development as they appear at this time. While this paper leaves much to be desired in achieving those ends, it is hoped that the framework drawn will be useful in assembling further information, past and future.

Past work on this type of engine, both abroad and in America, has been such that the results to date are of only preliminary importance. Barely a start has been made on the sizable improvements expected, and the field is almost incomparably rich in possibilities for creative endeavor, in the explanation and control of physical effects, in the conception of new mechanical designs, and in adaptation to propulsion of several types of aircraft. Perhaps the greatest challenge lies in the possibility of attaining high performance levels with an engine of such extreme simplicity and flexibility.

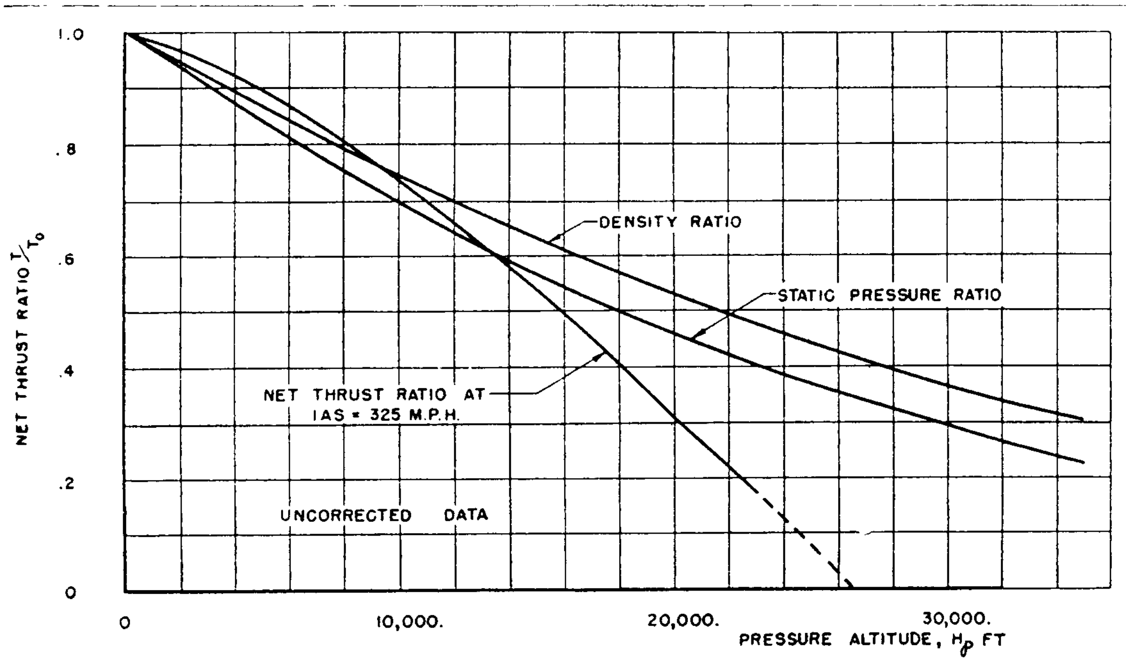


Fig. 17 – Altitude performance at constant indicated air speed for early experimental engine

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 1131

GAS-DYNAMIC INVESTIGATIONS OF THE PULSE-JET TUBE*

PART I

By F. Schultz-Grunow

FOREWORD

The gas-dynamic investigations of the pulse-jet tube conducted at the Technical High School at Aachen is presented in two parts. Part I was issued in June 1943 and Part II in August 1944; both par are presented here under one cover. The contents of each part is shown by the following brief summaries:

Part I - Influence of the form of the jet tube, of the effecti cross-sectional area of the valves, of leakiness in the valves, and of the speed of flight on the mode of operation of the pulse-jet tu and on the ratio of the amount of charge induced for the second cyc to the standard charge postulated for the first cycle.

Part II - Consideration of the sequence of pressure changes du combustion.

SUMMARY

Based upon a simplified representation of the mode of operation of the pulse-jet tube, the effect of the influences mentioned in the title were investigated and it will be shown that, for a jet tube with a form designed to be aerodynamically favorable, the ability to operate is at least questionable.

INTRODUCTION

The jet tube discussed herein is shown in its simplest cylindrical form in figure 1. Distributed over the cross section at the left-hand or inlet end of the tube are air valves that open automatically whenever the pressure within the tube is lower than that outside and permit only inward flow. Here also are located fuel-injection valves, which likewise open automatically at low pressure within the tube. The right-hand, or exhaust, end of the tube is open.

In a correctly proportioned jet tube, if a fuel-air mixture initially present in the inlet end of the tube in sufficient quantity is ignited, an automatically repeating working cycle begins, which consists of the explosion of the fresh charge, its exhaust, the sucking in behind it of a new mixture by the inertia of the exhausting gas column, and the automatic ignition of the new mixture, wherewith the process begins anew. The fuel-energy introduced is so transformed into heat and kinetic energy that exhaust occurs at a greater velocity than that of intake, whence a thrust results.

In particular, the processes of nonuniform movement in the tube will be discussed here on the basis of the law of propagation of gas-pressure waves of finite amplitude and in the simplest manner, namely, assuming adiabatic changes of state and further assuming that at the beginning of the working cycle there is present, adiabatically compressed relative to the surrounding air, a column of fresh charge, which suddenly expands. In accordance with the data given by Paul Schmidt, let the length of the column of fresh charge be one-seventh of the tube length, and the initial pressure 1.5 atmospheres above local atmospheric pressure.

These prescribed initial conditions largely predetermine the temporal course of variation of the excess pressure at the inlet cross section of the tube, from which the thrust results. The reader should therefore not expect to find in the following remarks a theoretical method of calculating the thrust; nor could this be expected in any case because the combustion process so completely eludes theoretical treatment that here experimentation alone is decisive. But what can be accomplished is the investigation of all the processes set in motion by the reverberating gas waves and the influence thereon of any structural alterations, combustion pressure, and flight speed. The criterion on which the effect of these factors will be more or less favorably adjusted will be, aside from the ability of the tube to operate, the increase achieved in the quantity of fresh charge sucked in and available at the end of a working cycle, inasmuch as a larger quantity will categorically possess a greater content of thermodynamically useful work. Data so obtained have been confirmed in experiments and have proven useful in the development of a jet-tube form of low air resistance. Moreover, such data facilitate the explanation of many phenomena observed in jet-tube work.

SYMBOLS

x distance, from inlet end toward exhaust end

t time, from beginning of a working cycle

R tube length

R' length of column of fresh charge at beginning of first working cycle

R'' length of column of fresh charge at beginning of second working cycle

u gas velocity

a velocity of sound

a_1 velocity of sound in local atmosphere

a_s velocity of sound at $u = 0$ or at stagnation point

$\xi = x/R$, nondimensional distance

$\tau = ta_1/R$, nondimensional time

$\xi' = R'/R$

$\xi'' = R''/R$

T period of one working cycle

p pressure

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*"Gasdynamische Untersuchungen am Verpuffungstrahlrohr." Inst. f. Mech., Tech. Hochschule Aachen (ZWB), Forschungsbericht Nr. 2015/1 u. 2.

p_1 local atmospheric pressure

ρ density

ρ_1 atmospheric density

$\kappa = c_p/c_v = 1.4$

M' quantity of fresh charge at beginning of first working cycle

M" quantity of fresh charge at beginning of second working cycle

F_R cross-sectional area of tube

F_E total effective cross-sectional area of air valves open for intake

$z = F_E/F_R$

FUNDAMENTALS

Method of investigation.

Making the assumption of one-dimensional movement along the axis of the tube, the pressure waves can be followed graphically in a system of coordinates of which the ordinate is time t and the abscissa the distance x traveled by the wave. An example of this representation of distance as a function of time is the graphic representation of a railway timetable in figure 2. The train leaves station A at time $t = 0$ and moves toward B with the constant speed u_1. At B it halts for Δt seconds and then proceeds further with the reduced speed u_2. The course of the line in figure 2 shows where the train is at a given time and when it will reach a given place. Its slope as measured from the t-axis represents the speed because $u = \frac{dx}{dt}$. The greater the deviation, the greater the speed. Zero slope from the t-axis indicates a state of rest; a leftward slope would indicate motion in the reverse direction.

In order to understand better the graphic methods to be used herein, reference may here be made to four reports in which the laws of propagation and reflection of pressure waves of finite amplitude in gases were investigated. (See reference 1, p. 322, and references 2 to 4.) According to these reports, there exists in a tube of uniform cross section the following relation between the velocity of sound a and the gas velocity u in an

outflowing (+x direction) wave, $du = -\frac{2}{\kappa - 1} da$

inflowing (-x direction) wave, $du = +\frac{2}{\kappa - 1} da$

or integrated
$$\Delta u = \mp \frac{2}{\kappa - 1} \Delta a_1 \tag{1}$$

These are linear relations, which permit a and u to be superimposed when opposing waves interact. These relations can also be so arranged that, with reference to the quantities defined by Riemann,

$$r = \frac{1}{2}\left(\frac{2}{\kappa - 1} a + u\right) \quad \text{and} \quad s = \frac{1}{2}\left(\frac{2}{\kappa - 1} a - u\right) \tag{2}$$

there belongs to each element of an outflowing wave an r value that does not change even when an inflowing wave is encountered, and to each inflowing element an s value that likewise does not change. One may further deduce that an outflowing-wave element does not influence the value of s along its path, nor an inflowing-wave element the value of r along its path, so if no encounter takes place, an outflowing wave has a constant s value and an inflowing wave a constant r value.

In the method to be employed, a pressure wave is approximated by a step-curve, as illustrated in figure 3. At each step, which shall hereinafter be designated a wave, a change occurs in the velocity of sound Δa and a change in the gas velocity Δu. If these changes in an outflowing wave are positive, there is a condensation wave; in the opposite case, a rarefaction wave. In order to distinguish them, condensation waves will be shown by solid lines and rarefaction waves by dashed lines. Figure 3 shows the wave propagation in the plane t,x and the whole wave as approximated by the wave elements at various times. The wave is understood to be caused by a variation of pressure operating at $x = 0$. In the diagram, the wave has been divided into equal increments Δu and Δa for the sake of simplicity, namely, $\Delta u = 1.1\ a_1$ and, consequently, from equation (1) with $\kappa = 1.4$, $\Delta a = 0.2\ a_1$. The arrows show the direction of the gas velocity u at each moment. The path of a gas particle in a t,x diagram is always shown as a finely dotted line; the deviation of this line from the t-axis represents u. The path shown is that of a gas particle that is at $x = 0$ when $t = 0$.

The movement and gas conditions in a wave are determined by a and u because pressure and density may be obtained from the adiabatic relations

$$\frac{p}{p_1} = \left(\frac{a}{a_1}\right)^{\frac{2\kappa}{\kappa-1}} \quad \text{and} \quad \frac{\rho}{\rho_1} = \left(\frac{a}{a_1}\right)^{\frac{2}{\kappa-1}} \tag{3}$$

The subscript 1 always indicates the atmospheric condition in which the jet tube is operating. [NACA comment: The author was able to neglect the differences in temperature between the working fluid and the outside atmosphere because of the dimensionless representation. The calculations were therefore made assuming that the working fluid in a state of rest has the same temperature as the outside atmosphere.]

The velocity of propagation of a wave w is composed of the velocity of sound a and the velocity of the gas u in which it travels. It is [NACA comment: In the general case, this relation is $w = u \pm a$.]

$$w = a + u \tag{4}$$

For the wave marked I in figure 3, $w = a_1 + 2\Delta a + 2\Delta u$ and specifically, as has been said, this velocity is represented in the plane t,x by $\tan \alpha$ (fig. 3). In drawing the wave plan, this slope is taken from a slope plan shown, on the scale of which the velocity corresponding to a given slope is indicated, both for the positive and, when required, for the negative direction of propagation.

Similarity.

In order to be bound as little as possible to concrete numerical values, the dimensionless coordinates ξ and τ will be used in place of x and t, so

$$\xi = \frac{x}{R} \quad \text{and} \quad \tau = \frac{a_1 t}{R} \tag{5}$$

where R is the tube length. The slope of a propagation line as measured from the τ-axis is then the velocity made dimensionless by divid-

ing by a_1. This mode of representation has the advantage that it is independent of a specific tube length or atmospheric condition (altitude). As a parameter expressing relative lengths, we now have simply

$$\xi' = \frac{R'}{R}$$

(R' = length of compressed column of fresh charge; R = tube length)

The initial pressure of the column of fresh charge, of course, constitutes a further parameter. Because this pressure has not yet been measured, as already stated, take as a reasonable assumption that it is 2.5 times as great as the pressure of the surrounding atmosphere. Then the velocity of sound in the compressed column of fresh charge according to equation (3) is [NACA comment: The ratio a/a_1 of 1.144 has apparently been chosen for convenience in subdivision into wave elements. It actually corresponds to a pressure ratio p/p_1 of 2.565, rather than 2.5.]

$$a = 1.144\ a_1$$

As also mentioned,

$$\xi' = \frac{R}{R'} = \frac{1}{7}$$

is taken as the normal case.

The absolute length of the tube does not appear here as a parameter, which is contrary to practical experience, because frictional and heat losses are disregarded.

As the final parameter, z is obtained, the ratio of the intake cross section F_E (effective flow area of the valves) to the tube cross section F_R

$$z = \frac{F_E}{F_R}$$

In using parameters, it is assumed that two tubes will behave similarly if all their parameter values are the same.

BOUNDARY CONDITIONS IN THE JET TUBE FOR WAVE REFLECTION

Wave reflection at the closed and at the open end of the tube, as well as at abrupt changes of cross section (a series that is considered as the approximate equivalent of a gradual change of cross section) have been previously treated (references 3 and 4).

Closed end of tube.

At the closed end of the tube, the gas velocity is always zero, whence it follows that a wave will be reflected with the same strength and with the same sign as in acoustics.

Open end of tube.

The pressure that exists at the open end of the tube during outward flow is that of the surrounding atmosphere, because the discharge is in the form of a jet; during inward flow there exists a sink flow and consequently a Bernoulli pressure decrease, which is determined by the Bernoulli equation

$$\frac{a^2}{\kappa - 1} + \frac{u^2}{2} = \frac{a_1^2}{\kappa - 1}$$

in which pressure is expressed by sound velocity. From this it follows, that so long as outflow exists, a wave will be reflected at full strength with opposite sign; but not so in the case of inflow, as a part of the wave then creates the Bernoulli pressure drop. With a decrease in the inflow, the Bernoulli pressure drop diminishes. This need not be further discussed because under the conditions assumed, so small a degree of inflow occurs at the open end of the tube that this pressure drop may be ignored.

Change in cross section.

At a change in cross section, the same quantity of fluid must enter at side A as that which leaves from side B.

$$\rho_A\ u_A\ F_A = \rho_B\ u_B\ F_B$$

The same applies to the energy:

$$\frac{u_A^2}{2} + \frac{a_A^2}{\kappa - 1} = \frac{u_B^2}{2} + \frac{a_B^2}{\kappa - 1} = \frac{a_s^2}{2} \tag{6}$$

These two conditions, which must apply to the passage of a wave through a change in cross section, may be most simply treated with the aid of a diagram, the so-called characteristics diagram, in which the gas velocity u is taken as the abscissa and the velocity of sound a as the ordinate. The diagram is preferably made dimensionless by dividing by a_1. Individual lines r = constant and s = constant, which from equation (2) are straight lines, are entered in the diagram. These are the characteristics, because at these lines discontinuities may occur, which in the present case are the waves. Actually, a value of r refers to an outflowing wave and a value of s to an inflowing wave. Also in the diagram (fig. 4) are curves of constant energy, which according to equation (6) are ellipses, and curves of constant mass velocity, which are hyperbolas in accordance with the equation

$$\frac{\rho u}{\rho_1 a_1} = \left(\frac{a}{a_1}\right)^{\frac{2}{\kappa-1}} \cdot \frac{u}{a_1} = \text{constant} \tag{7}$$

The passage of a condensation wave through a point of increase of cross section shall be observed, momentary states of which process before and after the reflection are shown in figure 5. The wave brings with it the gas state 6, namely, a_6 and u_6. It traverses state 2, defined by a_2 and u_2. At the point of increase of cross section, there occurs an increase of pressure to state 3. At the interaction of the wave with the tube enlargement, there arises from the wave 6 a wave 4, which proceeds forward, and a reflected wave 5. The states 6 and 5 are separated, according to figure 5, by an inward-traveling rarefaction wave and thus lie on a r = constant line in the a,u diagram; the states 4 and 3 are separated by an outward-traveling condensation wave and thus lie on a s = constant line. States 6 and 3 are known and hence also the r- and s-lines on which states 5 and 4, respectively, lie. States 5 and 4 are separated by the point of change of cross section and thus lie also upon an ellipse. Furthermore there exists between 5 and 4 a difference of mass velocity $\Delta(\rho u)$ determined by the difference in cross section ΔF in accordance with

$$\Delta(\rho u) = \rho_5\ u_5 - \rho_4\ u_4$$

and

$$\rho_5 \, u_5 \, F_5 = \rho_4 \, u_4 \, F_4$$

in which F is the cross-sectional area. Hence

$$\Delta(\rho u) = \rho_5 \, u_5 \, \frac{\Delta F}{F_4} \qquad (8)$$

Thus there is only one ellipse on which 5 and 4 may lie, namely, the one whose points of intersection with the given r- and s-lines have the prescribed difference of mass velocity. The ellipse must be found by trial. In figure 6 the reflection is shown in a t,x diagram. The wave reflected is a rarefaction wave because a_4 $a_6 > a_5$ and $u_5 > u_6$. In the case of a reduction in cross section, the reflected wave is a condensation wave.

Reflection at the valves.

The valves at the intake end behave like a solid wall, so long as there is an excess of pressure against their inner sides. But when the pressure is less than that outside they are open; in this case reflection occurs as at a partly open tube end. Inasmuch as the valves are spring-operated, their openings increase with decreasing inside pressure to a maximum area F_E determined by their mechanical construction. For the sake of simplicity, this spring action shall be disregarded and it shall be assumed that whenever the inside pressure is less than that outside, the valves are fully open. Let F_E be the effective cross-sectional area of the openings, in which the effect of friction and the vena-contracta loss have already been allowed for. The Bernoulli equation (equation (6)) for uniform flow may be applied to the flow through the valves on account of their short flow length. The flow conditions in the valves thus lie in the a,u diagram along an ellipse that intersects the a-axis (that is, the line $u = 0$) at a point corresponding to the value of the sonic velocity at the stagnation point a_s, provided that by a means of suitable fairing (total head scoop) ahead of all the valves, the impact pressure corresponding to the speed of flight is made to operate against their outer sides. Let the state of the gas and condition of motion occurring at the narrowest valve cross section be denoted by 4. Neglecting the shock loss, it is assumed that on the outflow side of the valves, that is, the inner side, there is a uniform distribution of velocity over the tube cross section F_R, and hence

$$\rho_4 \, u_4 \, F_4 = \rho_3 \, u_3 \, F_3$$

and

$$\Delta(\rho u) = \rho_4 \, u_4 - \rho_3 \, u_3 = \rho_4 \, u_4 \left(1 - \frac{F_E}{F_R}\right) \qquad (9)$$

in which 3 denotes the state of the gas and condition of motion in the tube immediately next to the valves. Because, with the construction chosen, the outflow from the valves occurs without any regaining of pressure, $a_3 = a_4$, and states 3 and 4 lie on a horizontal line in the a,u diagram. For any mass velocity $\rho_4 \, u_4$, the mass velocity difference $\Delta(\rho u)$ may be computed from equation (9) because F_E/F_R is given by the dimensions of the tube.

Now if a gas and motion condition 1 exists at the discharge side of the valves, through which a rarefaction wave travels bringing with it condition 2 (fig. 7), then a wave will be reflected at the valves that will produce state 3 at the discharge side and state 4 in the narrowest valve cross section. States 1 and 2, because they are separated by an inward-traveling wave, lie on a r = constant line in the a,u diagram (fig. 3); states 2 and 3, being separated by an outward-traveling wave, lie on a s = constant line. The required conditions 3 and 4 lie on that horizontal straight line on which 3 and 4 will have the mass velocity difference $\Delta(\rho u)$ prescribed by equation (9). In figure 8 may be seen four possible cases of the reflection, according to the magnitudes of $\Delta(\rho u)$ and F_E/F_R. At a ratio F_E/F_R in the neighborhood of unity, to which the states 3^1 and 4^1 in figure 8 correspond, a condensation wave will be reflected. Then there is a smaller F_E/F_R value at which no wave will be reflected, states 3^2 and 2 coinciding. At a still smaller F_E/F_R value, a rarefaction wave will be reflected, because 3^3 has smaller state values than 2. At a still smaller F_E/F_R value, there occurs the limiting case in which $u = a$ in the valve cross section (the flow velocity u equals the velocity of sound a); this is condition 4^4. The corresponding state 3^4 is the point of intersection of the s = constant line through 2 with that hyperbola which has the required increment of mass velocity $\Delta(\rho u)$ relative to the hyperbola that passes through 4^4 and is tangent to the ellipse. In figure 9 the reflection in the third case is shown in a t,x diagram.

MODE OF OPERATION OF THE JET TUBE AT REST

Cylindrical Tube

The mode of operation of the cylindrical tube is evident from the wave diagrams in figures 10 and 11, which are drawn for $F_E/F_R = 0.2$ and 0.4, respectively. A part of the excess pressure of the column of fresh charge travels toward the right as the condensation shock wave a, which is so small that the change of state in it may be regarded as adiabatic. The other part travels leftward as a rarefaction wave; the initial difference of velocities of sound $\Delta a = a - a_1 = 0.144a_1$ (see p. 6) distributes itself one-half to each of the two waves. The rarefaction wave is shown as divided into two elementary waves b and c, in order to allow for the lesser velocity of propagation of later wave elements than that of earlier ones. A further subdivision proves to be unnecessary. The rarefaction wave is reflected at the left end of the tube as from a solid wall because the valves are closed. Finally, therefore shock a and the two waves b and c are traveling toward the right. The states of the gas and conditions of motion created by the shock and the waves are enumerated by the figures in circles, for which the state values, namely, velocity of sound a/a_1 and gas velocity u/a_1, may be found in table I. Positive gas velocity signifies velocity directed toward the right. It may be seen from the table that behind the last rarefaction wave traveling to the right, atmospheric conditions are again attained.

At the right-hand or open end of the tube, the shock is reflected as a rarefaction wave, which is likewise divided into two elementary waves d and e. The rarefaction waves are here reflected at the open end as condensation waves, which soon combine themselves into the shock h.

The rarefaction wave d creates condition 12, for which table I shows pressure lower than the outside atmosphere. As soon as this state reaches the left-hand end of the tube, the valves open and at the moment T_A fresh charge begins to flow in. Consequently, the finely dotted line representing propagation of the boundary of the fresh charge begins here. The intake process is strengthened by the subsequent rarefaction wave e.

Because of the shock h, which now arrives, the fresh-charge front is forced not merely to stop but to reverse its motion, whereby

the fresh charge is compressed. As soon as the shock strikes the valves, the intake period is ended and we have at this moment τ_E a compressed column of fresh charge of the length ξ'' in the tube. It is assumed that at the moment τ_E the explosion of the column of fresh charge occurs, so the length ξ'' then attained has the same significance for the second working cycle as ξ' for the first. Because it turns out that $\xi'' < \xi'$, it must be concluded that the jet tube could only operate if $F_E/F_R > 0.4$. The explanation of the fact that in reality it operates even when $F_E/F_R = 0.2$ lies in the choice of prescribed conditions. In spite of this inconsistence with reality, figures 10 and 11 show a number of noteworthy particulars that do correspond to reality.

1. The intake process is set in motion by the rarefaction waves d and e into which the condensation shock wave a is transformed without loss of strength by its reflection at the right-hand end of the tube.

2. It appears that the condensation shock wave h, into which the rarefaction waves b and c are transformed by reflection at the right-hand end of the tube, plays a part in the ignition of the fresh charge.

3. At small F_E/F_R ratios of the order of magnitude of 0.3 or less, the rarefaction waves d and e, which start the intake process, are reflected as the waves i and k of the same kind and, in fact, produce a gas velocity directed toward the left so the working cycle ends with an inflow at the exhaust end. In actual fact there has been observed with the Argus tube an inflow at the exhaust end preceding the exhaust of the next cycle when $F_E/F_R = 0.3$.

At $F_E/F_R = 0.4$, waves i and k no longer set in motion a negative velocity; only k is a rarefaction wave. It is thus to be expected that, in agreement with the cited observation, at $F_E/F_R > 0.4$ this inflow at the exhaust end does not occur.

EFFECTS OF CONSTRUCTIONAL ALTERATIONS

Jet Tube at Rest

Influence of length ξ', tube length remaining the same.

The greater ξ', the longer is the path of the rarefaction waves b and c (figs. 10 and 11), so much later is the shock h formed from them and so much longer is the intake period $\tau_E - \tau_A$ and the duration of the working cycle.

The effect on the duration of the working cycle is negligible, amounting to 5 percent. The increased intake period results in an increased quantity of fresh charge M" at the end of the working cycle. With an increase of ξ' from 0.1145 to 0.191, that is, an increase of 66 percent, there is obtained

F_E/F_R	Increase of M" (percent)
0.4	20
.6	30
.8	30

Influence of tube length R, absolute length of R' remaining the same.

In the limiting case of very weak waves, one would expect, according to the laws of acoustics, an increase in the duration of the working cycle proportional to R. In this case it slowly increases as much as 5 percent because as R increases the rarefaction waves b and c shift nearer to the shock a, thereby reducing the extent of the regions through which the waves d and e move more slowly.

M" increases with increasing R as follows:

F_E/F_R	R (m)	M"
0.4	5.25	0.6
	7	.7
	8.75	.8
0.6	5.25	0.9
	7	.9
	8.75	1.0
0.8	5.25	1.0
	7	1.0
	8.75	1.1

According to this table, the results of lengthening the tube are favorable; of course there is, for reasons not here discussed (heat leakage) a practical limit of about R = 3.5 meters.

Influence of change in valve cross section F_E.

Figure 12 shows the increase of M" with increasing F_E/F_R on the basis of figures 10 and 11 and similar figures.

Influence of tube shape (tube of varying cross section).

The values $\xi' = 1/7$ and $F_E/F_R = 0.4$ were taken as a basis. The tube forms investigated with discontinuous change of cross section, in which each change from left to right always amounts to 50 percent of the preceding section, are shown in figure 13. Beside them to the right are given the tube forms with continuous change of cross section, to which the investigated stepped forms may be regarded as approximating. Less accurately than before, a rarefaction wave is now represented with one instead of two rarefaction lines, for at each change of cross section each wave gives rise to two, whereby the investigation becomes very complicated.

Form A corresponds to the tube already investigated in figure 11, which is to serve as a basis for comparison.

(a) Reduction of cross section at ξ' (form B):

The fresh-charge front is assumed to lie a little to the left of the constriction. This form corresponds approximately to the cylindrical Argus tube with enlarged combustion chamber.

The applicable wave diagram is given in figure 14. As a result of the constriction, the condensation shock wave a is stronger and the rarefaction wave b is weaker than with form A (fig. 11). Consequently, the rarefaction wave c resulting from the reflection of a is stronger than the corresponding wave in figure 11. The result is smaller velocities of propagation for the subsequent condensation waves f and g and consequently a substantially longer intake

period $\tau_E - \tau_A$ than in the case of form A, a substantially larger column of fresh charge being thus drawn in.

In this connection it must be remembered that the returning waves e, f, and g are weakened at the point of change of cross section. The fresh charge will therefore be sucked in with a weaker vacuum than in the case of figure 11. But if the quantity M" newly drawn in is compared, it is found that nevertheless

$$\frac{M''_B}{M''_A} = \frac{0.823}{0.51} = 1.6$$

that is, in spite of the weaker suction the intake period is sufficiently lengthened that form B yields a 60-percent increase in charge.

Through the weakening of the subsequent wave of condensation, the ignition of the new column of fresh charge is made doubtful. Of course, the condensation shock wave creates a higher absolute pressure than in form A. It shall not be attempted to decide which is more decisive for ignition, the absolute pressure or the pressure ratio.

The optimal constriction is that at which the exhaust occurs with a velocity equal to that of sound in the surrounding atmosphere.

(b) Reduction of cross section at center of tube length (form C):

In figure 15 waves d and e, reflected at the constriction, cross the returning waves f and g too late to produce as long an intake period $\tau_E - \tau_A$ as in figure 14. Besides this, there occurs a still smaller pressure drop across the valves, so now

$$\frac{M''_C}{M''_A} = \frac{0.71}{0.51} = 1.39$$

the improvement over form A still amounts to 39 percent.

The pressure ratio in the condensation shock wave h, which produces the ignition, is just a little smaller than with form B; however, the absolute value of the compressing pressure is greater.

(c) Reduction of cross section at exhaust end (form D):

From figures 10 and 11 it may be seen that the automatic operation of the jet tube is based on the circumstance that at the open end a condensation wave is reflected as a rarefaction wave, and a rarefaction wave as a condensation wave, namely, at full strength because at the exhaust end only outflow takes place. Consequently, the first to strike the valves is the rarefaction wave, which starts the intake period, after which the igniting condensation shock wave h reaches them.

But if now the reduction of cross section is located at the exhaust end, this means that there is a half-open tube end at which, as closer investigation shows (fig. 8), the condensation wave is reflected as such with lessened strength and likewise the rarefaction wave is reflected as such with lessened strength. In this case, as shown in figure 16, the condensation and rarefaction waves strike the valves in reversed sequence and the tube does not function.

(d) Two reductions of cross section (form E):

Here the lengthening of the intake period determined for forms B and C is effected by two constrictions of 50 percent each. A longer intake period $\tau_E - \tau_A$ but a lesser intake-producing pressure difference is thus obtained (fig. 17). Likewise, the pressure ratio in the igniting condensation shock wave h is the smallest of those for all tube forms. In practice the absolute compression pressure is of equal magnitude to that obtained with form C. The newly induced quantity of fresh charge M''_E compares with form A as follows:

$$\frac{M''_E}{M''_A} = \frac{0.856}{0.51} = 1.68$$

This tube form gives the greatest increase in charge over the simple tube of form A. This result is not directly comparable with the results for forms B and C, because in form E the tube cross section is reduced by 75 percent but in the other forms only by 50 percent.

(e) Increase of cross section at open end of tube (form F):

The adverse effect of a reduction of cross section at the exhaust end having been shown, the question arises, whether an increase of cross section at the end of the tube is desirable.

In figure 18 it may be seen that the first waves traveling to the right, waves a and c, are reflected at the point of increase of cross section with reversed signs as waves d and e and that by these waves a short intake process and an explosion are effected, the fresh charge column ξ'' being drawn in during the period $\tau_E - \tau_A$ and exploded.

At the open end of the tube, the original waves a and c are reflected with reversed signs as in the normal case, giving rise to the rarefaction wave f and the subsequent condensation wave g. Although these waves encounter the pressure wave arising from the ignition of the column of fresh charge ξ'', they likewise effect an intake process and an explosion, the fresh charge column ξ''' being drawn in during the period τ'_A to τ'_E.

Thus the remarkable circumstance exists that during one working cycle of the tube, intake and explosion occur twice. Comparing the intake quantities M" and M''' contained in the columns of fresh charge ξ'' and ξ''' with the normal case of the cylindrical tube in figure 11, it is found that during the first period, τ_A to τ_E, 45 percent less is drawn in but in the second period, τ'_A to τ'_E, 40 percent more is drawn in; the charge is therefore almost doubled by this increase in the cross section. Of course the doubled frequency of opening of the valves must be regarded as unfavorable to their length of life.

INVESTIGATION OF A STREAMLINED TUBE

(Argus VSR 9z, Drawings II 11039 and II 11036)

This tube has a form selected as aerodynamically advantageous. Its realization was very earnestly desired as a means of reducing the air resistance, which with the cylindrical jet tube is excessive. It will be shown that from a gas-dynamic aspect, the ability of this cigar-shaped tube to operate is very doubtful. Tests have meanwhile confirmed this: The tube proved itself incapable of operation on the test stand. The investigation of this tube affords a practical application of the arguments set forth.

In accordance with the method of substituting discontinuous for continuous changes, the longitudinal cross section of the tube is approximated by a step-curve with three steps. At each step the ratio of the smaller to the larger cross section is 0.63.

Because it is a question of an approximation, there remains a certain leeway as to how the real form shall be approximated by the step-curve. For practical reasons, two step-curves representing two extremes were chosen; one (figs. 19 and 20) in which the step-curve lies on the outside of the real shape; and another (fig. 21) in which it lies inside the real shape.

With these two step-curves two limiting cases between which the real shape lies may be investigated. The investigation of the limiting cases has this advantage, that from their different gas-dynamic behavior, it can be deduced how the given streamline form may be improved by slight alterations. This process is particularly facilitated by figure 22, where for each step-curve a mean continuous curve is given, between which the actual shape lies. The investigation will show which of these limiting continuous curves must be approached in order to improve the gas dynamics of the streamlined tube as far as possible, or, for that matter, to make the streamlined tube operable at all, because it should be pointed out at once that the operability of the streamlined form as given is open to doubt because of the marked constriction at the end of the tube.

In order to make possible a comparison between this investigation and the results obtained earlier for the cylindrical tube of figure 11, the ratio of the effective inlet cross-sectional area to the tube cross-sectional area at the valve plate shall be disregarded and as the parameter the ratio of the effective inlet cross-sectional area to the maximum cross-sectional area of the tube, namely, $F_E/F_R = 0.4$, shall be taken. As the initial condition at the beginning of the working cycle, it is again assumed that the tube is charged for 1/7 of its length with fresh charge, which is under an initial pressure of 2.5 times atmospheric pressure, corresponding to a ratio of sonic velocities of 1.144.

In order to keep the investigation from becoming unduly complicated, in figures 19 and 21 each rarefaction wave will again be represented by a single rarefaction line. In figure 20 the rarefaction waves for the case corresponding to figure 19 are shown in better approximation by means of two lines, as a check on whether the approximation by one line is sufficiently exact.

Let the step-curve shown in figure 19 now be investigated. Here the last abrupt change in cross section takes place at the end of the tube. The condensation shock wave in passing through the changes of cross section gives rise to condensation waves traveling in the opposite direction; at the same time it is so strengthened that the gas leaves the tube with a pressure greater than atmospheric and at the velocity of sound (condition 8).

The conditions at the end of the tube are now such that for all practical purposes the condensation shock wave is not reflected at all. This is of serious consequence for the operation of the tube because the automatic operation of the tube depends directly on the reflection of the condensation shock wave from the end of the tube as a rarefaction wave and on the opening of the inlet valves by this returning rarefaction wave. As this reflected rarefaction wave is now lacking, the valves are not opened at all and the tube thus cannot function.

Similar considerations apply to the subsequent rarefaction wave c. It is reflected as a rarefaction wave at the changes of cross section within the tube, simultaneously strengthened by entry into the narrower sections of the tube, and not reflected at the open end of the tube so the reflected condensation wave, which in its return would ignite the new charge, likewise does not arise.

The waves reflected at the changes in cross section within the tube are also not capable of maintaining the automatic operation of the tube, for the waves reflected are always of opposite sign to those required for operation, that is, a rarefaction wave instead of a condensation wave and vice versa. Furthermore, the reflected waves eventually so overtake each other and cancel that after only one cycle of reverberation the atmospheric condition of rest is restored.

By means of figure 20, it is now inquired whether this negative result is due to the rough approximation of the rarefaction wave by a single rarefaction line b. In figure 20 therefore, the rarefaction wave is represented more exactly by means of two rarefaction lines, as originally done for the cylindrical tube. This subdivision certainly has no influence on the condensation shock wave a and it is therefore exactly as in figure 19, reflected at the exhaust end of the tube not as the required rarefaction wave but as a very weak condensation wave, which in practice may be disregarded.

Now it could be possible that by reflection at the intermediate changes of cross section, waves might arise that would support the automatic operation of the tube. It is found, however, that even with this more accurate representation the reflected waves d to i create no subatmospheric pressure at the intake and consequently the valves are not opened. The lowest pressure that occurs at the inlet end is that of the atmosphere, exactly as in figure 19; the two figures likewise agree in respect to minimum pressure. The simplified conception of the form of the rarefaction wave is therefore not the cause of the negative result.

The investigation of the second step-curve is illustrated in figure 21. At the exhaust end there is not a point of choking but a cylindrical portion of the tube; the individual drops in cross section are shifted toward the left as compared to the first step-curve. The investigation gives significantly better results in this case. The condensation shock is again strengthened in passing through the drops in cross section and now, because the exhaust portion of the tube is cylindrical, it is reflected as a rarefaction wave. The strengthening of the condensation shock wave in passing through the drops in cross section produces a strong reflected rarefaction wave and an outward flow of the gas from the tube at the velocity of sound. This reflected rarefaction wave, due to its strength, produces upon arrival at the inlet end a strong intake action having a gas velocity of 0.2 and a subatmospheric pressure corresponding to a velocity of sound equal to 0.976 (condition 39, table VIII). A comparison with the cylindrical tube shows that the corresponding values for the cylindrical tube are: gas velocity, 0.54; velocity of sound, 0.924. The intake conditions for the second step-curve are therefore less favorable than for the cylindrical tube.

The subsequent rarefaction wave c, being likewise strengthened at the drops in cross section, is reflected as a relatively strong condensation wave l from the exhaust end of the tube; but the strength of this wave is disadvantageous here, as its strength involves a high velocity of propagation and consequently it does not arrive at the intake area late enough after the rarefaction wave to allow enough fresh charge to be drawn in. Because of this, in comparison with the cylindrical tube only 74 percent as great a quantity of fresh charge is drawn in.

The waves reflected from the points of change of cross section, as these points are encountered by the original waves a and c, exhibit no effect upon the mode of operation of the tube, because they cancel each other rather quickly, certainly before the rarefaction wave arising from the reflection of the condensation shock wave at the tube mouth passes back through the constrictions.

The step-curves may be regarded as approximations of mean curves that depart somewhat from the given longitudinal section: the one curve (outer step-curve) in the sense that it is more rounded-in at the exhaust end; the other curve (inner step-curve) in the sense that it is more elongated and cylindrical at the exhaust end. These mean curves are compared with the actual tube form in figure 22. On the basis of the results obtained, the more rounded-in form must be considered out of the question. With this form the automatic operation of the tube must be considered impossible. The more nearly cylindrical form ought to be operable, especially in flight, where the intake conditions are more favorable than in the stationary test setup that has been considered.

In any case the comparison of the three tube forms shown in figure 22 shows how sensitive is the automatic operation of the tube to the influence of the shape of its exhaust end, because the exhaust ends of the three shapes actually differ relatively little. Unquestionably, the end of the tube must be as cylindrical in form as possible. In other words, it may also be said that although the cigar-shaped streamlined form shows itself in tests to be unsuitable for automatic tube operation, it may be improved by slight changes at the exhaust end of the tube, such as by the addition of a cylindrical portion to the tube; of course, a gradual transition from cigar to cylinder shape must be provided.

MODE OF OPERATION OF JET TUBE IN FLIGHT

General considerations.

In the case of the tube in flight, the simplified consideration in terms of adiabatic processes need be modified by only one new condition, namely, that the outside pressure on the inlet valves is the impact pressure. It is presupposed that the full effect of the impact pressure is directed against all valves by suitable fairing (total head scoop).

The impact pressure is computed from the Bernoulli equation for a compressible flow

$$\frac{a_1^2}{\kappa - 1} + \frac{u^2}{2} = \frac{a_s^2}{\kappa - 1}$$

in which a_s = velocity of sound at stagnation point and a_1 = velocity of sound in the undisturbed atmosphere. As before, all velocities are made dimensionless by dividing by the velocity of sound a_1. Be it particularly noted that a_1 refers not to ground level but to the altitude of flight at the moment in question.

A further assumption is that even under the full effect of the impact pressure, the valves maintain gas-tight closure and that, upon the arrival of the reflected rarefaction wave at the inlet end of the tube, they at once completely open. This assumption does not exactly correspond to the facts because the valve flaps must be held forward with spring pressure in order to remain closed against the force of the impact pressure. Hence, they will be opened by the wave of lower pressure only so wide as corresponds to the additional load imposed by the lower pressure, which at various flight speeds will naturally vary in proportion to the impact pressure. The investigation of this influence of the strength of the valve springs on the opening of the valves, however, will be postponed until later.

In treating the wave reflection at the valves, it must be noted that the velocity of sound of the stagnation point a_s and not that of the atmosphere a_1 now applies to the ellipse in figure 8 u = 0.

Influence of flight speed on amount of charge.

The investigation will be carried out for

$$\frac{F_E}{F_R} = 0.2 \quad \text{and} \quad \frac{F_E}{F_R} = 0.4$$

again with $\xi' = 1/7$ and $a'/a_1 = 1.144$.

The individual wave diagrams, which were drawn for flight speeds of 0.3, 0.45, 0.55, 0.78, and 1.00 times the velocity of sound, are not given in the appendix; simply the results in figures 23 and 24 are given, first as the ratio of the quantity of fresh charge M" newly drawn in to the quantity M' originally present plotted against flight speed and then as the ratio of M" at the given flight speed to the quantity M_0'' corresponding to zero flight speed plotted against flight speed.

As the impact pressure increases, a point is reached at which the gas velocity in the valve cross section F_E is equal to the speed of sound. With a further increase in impact pressure this critical state persists, because with the chosen valve construction no supersonic speed can arise. Nevertheless the quantity of inflowing gas continues to increase because at a pressure ratio above the critical the gas quantity depends solely on the state of the gas on the pressure side of the valves. Thus, rarefaction waves arriving from the open end of the tube and striking the valves after the critical condition has been reached have no further effect on the quantity of gas flowing in.

The presentation of the results in figures 23 and 24 shows that with increasing flight speed the quantity of new charge flowing in increases slowly at first and then more and more markedly. Figure 23 also shows the obvious fact that with $F_E/F_R = 0.4$ the inflow is consistently greater than with $F_E/F_R = 0.2$. In the terms of figure 24, the percentage increase is the same whether $F_E/F_R = 0.2$ or 0.4.

Influence of leakiness of valves.

Figure 10 shows that when $F_E/F_R = 0.2$, the minimum pressure occurring at the inner side of the valves corresponds to the ratio $a/a_1 = 0.896$. The pressure corresponding to this is

$$\left(\frac{a}{a_1}\right)^{2\kappa/(\kappa-1)} = 0.464;$$

that is, the ratio of the absolute pressure to the atmospheric pressure. The pressure difference at the valves amounts thus to 54 percent of the atmospheric pressure.

On the other hand, for a tube flying at 0.64 of the velocity of sound, the ratio of impact pressure to atmospheric pressure = 1.32. Now it has been seen from the preceding investigations that atmospheric pressure is attained behind the pressure wave as it travels out of the tube. Between the explosion and the subsequent intake there is atmospheric pressure on the inner side of the valves, so that a pressure difference at the valves of 32 percent of atmospheric pressure then exists. During the intake period, a maximum pressure difference of 54 + 32 = 86 percent of atmospheric pressure exists.

The fixed tube and the tube in flight shall now be compared under equal atmospheric conditions. If at rest, the valves open to the degree $F_E/F_R = 0.2$ at the maximum, that is 54 percent pressure difference; then in flight they open at the same low pressure approximately to the degree $F_E/F_R = 0.2\,\frac{86}{54} = 0.32$. During the interval in which atmospheric pressure exists on the inner side of

the valves at rest, the valves are closed but in flight they are open approximately to the degree $F_E/F_R = 0.2 \frac{0.32}{0.54} = 0.12$. This opening is designated the leakiness. Relative to the maximum opening it amounts to $\frac{0.12}{0.32} = 37$ percent.

It is thus seen that at sufficiently high flight speeds, the valves are constantly open excepting of course at the time of explosion. It shall now be investigated how the jet tube reacts to this condition.

Inasmuch as no exact data on valve opening exist, the effects of a leakiness of 50 percent and of 100 percent shall be investigated with the assumptions of maximum attained valve opening $F_E/F_R = 0.2$ and flight speed of 0.64 of the velocity of sound.

The investigation is made by means of figures 25 and 26. The only difference from the earlier case shown in figure 10 is that from the instant τ', when the outward traveling rarefaction wave leaves the valves, fresh charge begins to flow in through a cross-sectional area $\Delta F_E/F_R = 0.1$ (fig. 25) or 0.2 (fig. 26). The boundary of the inflowing fresh charge is again shown by a finely dotted line. For the succeeding intake period it is assumed that the mechanical limit to the valve-opening area is $F_E/F_R = 0.2$ and that it is attained. In reality, of course, the opening area depends upon the pressure head, which varies during the intake period; it would scarcely be worth the effort, however, to take account of the consequent alteration from moment to moment of the effective inlet area.

If the cases of figures 25 and 26 are compared as to the quantities of fresh charge available for the second working cycle, that is, the quantities present at the moment τ_E, figure 27 is obtained, which shows the increase with increasing leakiness. From this figure it can be seen that the increase of the quantity of fresh charge due to a slight leakiness of the valves is quite advantageous but that greater leakiness results in a too great increase in the charge, in the sense that a sufficiently rapid spread of ignition throughout the gas becomes doubtful and also that too great a cooling of the tube may be expected.

In order to evaluate the operability of the tube with valve leakiness, the strength of the returning wave must be considered. The investigation shows that the leakiness does not affect the low pressure created at the valves by the returning rarefaction wave, the suction effect thus remaining the same but that the returning condensation shock wave is somewhat weakened. The decrease of the pressure ratio (ratio of pressure after to pressure before the shock) produced by the compression shock wave is shown in figure 28. The reason for this effect is, that whereas the outward-traveling condensation shock wave a is unaffected, consequently the rarefaction waves d and e arising from it by reflection are unaffected; on the other hand, the outward traveling rarefaction waves b and c, and also the condensation shock wave f arising from them by reflection, are weaker because the gas condition 7 behind the outward traveling wave possesses a higher pressure than it would at zero leakiness. The decrease of the pressure ratio in the shock wave f is 20 percent at 100 percent leakiness, according to figure 28.

Furthermore, it is found that the rarefaction waves g and h reflected from the valves become stronger with increasing leakiness without, however, giving rise to any negative gas velocities (that is, inflow at the exhaust end near the end of the working cycle).

A limited leakiness has an advantageous effect inasmuch as it increases the weight of new charge induced and only slightly affects the waves. Greater leakiness results in a weakening of the returning compression shock and therefore has an unfavorable effect on the ignition, as does also the cooling of the tube and the less rapid spread of ignition throughout a larger as compared to a smaller quantity of fresh charge.

INFLUENCE OF COMBUSTION PRESSURE

Although the influence of the combustion pressure is of less practical significance because its magnitude cannot be predetermined, for instance by the choice of fuels, it is nevertheless of interest inasmuch as according to the research of Busemann (reference 5) the best yield of work from a thermodynamic viewpoint is secured from a jet tube with combustion at constant volume and the question arises, whether the jet tube is operable with this kind of combustion.

Combustion at constant volume is characterized by a higher combustion pressure than normally occurs in the pulse-jet tube. It will therefore be possible to decide the question approximately by investigating the influence of the combustion pressure.

The results of the investigation are shown in figure 29. According to this figure, the degree of charge and the pressure ratio in the reflected compression shock at first increases with increasing combustion pressure p then reaches a maximum in the neighborhood of which the curves are level and thereafter decrease, so at too high a combustion pressure, in this case namely at $p/p_1 > 7$, the operability of the tube becomes questionable.

The reason is that with increasing combustion pressure a point is reached at which the exhaust flow reaches the velocity of sound and beyond that point the outward-traveling explosion wave is no longer reflected at full strength as a rarefaction wave because the reflected wave has zero net velocity of propagation against the gas flowing out with the velocity of sound. This reflected wave remains at the mouth of the tube and is only disposed of by the subsequent arrival of the rarefaction waves b and c, as a result of which these waves are in part reflected as rarefaction waves and not condensation waves as they should be; consequently the reflected compression shock is weaker.

From equation (1), the combustion pressure at which the exhaust will reach the speed of sound can easily be computed. The sound velocity attained at the exhaust end is then that of the surrounding atmosphere. Letting a_v be the sound velocity corresponding to the combustion pressure, then on the one hand

$$\Delta a = a_v - a_1$$

on the other hand

$$u = \frac{2}{\kappa - 1} \Delta a = a_1$$

and therefore

$$a_v = \frac{\kappa - 1}{2} a_1 + a_1 = 1.2 a_1$$

$$\frac{a_v}{a_1} = 1.2$$

or

$$\frac{p}{p_1} = 3.6$$

But the pressure occurring in the jet tube is actually $p/p_1 = 2.5$.

INFLUENCE OF TUBE FORM ON VARIATION OF PRESSURE WITH

TIME AT VARIOUS TUBE CROSS SECTIONS

In order to make a comparison with measurements of the sequence of pressures taken through taps in the tube walls, the theoretical sequences of pressures at the points $\xi = 0.4$ and $\xi = 0.6$, that is at distances of 0.4 and 0.6 of the tube length from the inlet end, were derived from the wave diagrams of figures 11, 14, and 15, corresponding to the tube forms A, B, and C of figure 13. The results are shown in figures 30, 31, and 32 for one working cycle. These figures show the ratio of tube pressure to atmospheric pressure p_1 plotted against dimensionless time T. The arrows above the waves indicate their direction of propagation in the tube.

In comparing the cylindrical tube, form A (fig. 30), with that having the enlarged combustion chamber, form B (fig. 31), it is noteworthy that the pressure varies less widely for form B than for form A. This fact had been observed in the experimental measurements and had then led to the opinion that a less vigorous combustion took place in form B. But it is now apparent that the cause lies entirely in the tube form. As to form C, figure 32 shows that a constriction half-way along the tube gives rise to additional pressure peaks. In this case, the outward-traveling and inward-traveling waves are not separated by a time interval as in the preceding tube forms, a circumstance that makes the evaluation of experimental measurements more difficult.

TABLE I

STATE VALUES FOR FIGURES 10 AND 11

$F_E/F_R = 0.2$ $F_E/F_R = 0.4$

Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1.000	0	1	1.000	0
2	1.072	.36	2	1.072	.36
3	1.036	.18	3	1.036	.18
4	1.144	0	4	1.144	0
5	1.108	.18	5	1.108	.18
6	1.072	0	6	1.072	0
7	1.076	.54	7	1.036	.54
8	1.000	.72	8	1.000	.72
9	1.000	.36	9	1.000	.36
10	.964	.54	10	.964	.14
11	1.000	.36	11	1.000	.72
12	.964	.18	12	.964	.39
13	.928	.36	13	.928	.33
14	.964	.18	14	.964	.18
15	.954	.13	15	.970	.21
16	.918	.31	16	.974	.39
17	.966	.20	17	.922	.34
18	.990	-.03	18	1.006	.03
19	.968	-.16	19	.998	.02
20	.954	.37	20	.970	.52
21	.910	.910	21	.922	.3[illegible]

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2. Sauer, R.: Ing.- Archiv, Bd. XIII, 1942, p. 79.

3. Schultz-Grunow, F.: Forsch. Ing.- Wes., Bd. 13, Heft 3, 1942, p. 125.

4. Schultz-Grunow, F.: Ing - Archiv, Bd. XIV, Heft 1, 1943, pp. 21-29. ("Non-Steady, Spherically-Symmetrical Gas Motion and Non-Steady Gas Flow in Nozzles and Diffusors." British R.T.P. Trans. No. 2071, Ministry Aircraft Prod.)

5. Busemann, A.: FB 530, ZWB.

TABLE II - STATE VALUES FOR FIGURE 14

Form B

Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1	0	14	0.911	0.449
2	1.141	0	15	[illegible]	.200
3	1.080	.445	16	.908	.530
4	1.103	.205	17	.904	.730
5	1.088	0	18	.987	.255
6	1.042	.100	19	1.0[illegible]	.017
7	1.04	.200	20	1.007	.075
8	.911	.645	21	1.01	.087
9	1	.080			
10	1	.400			
11	1.022	0			
12	1.015	.075			
13	1.015	.070			

TABLE III - STATE VALUES FOR FIGURES 15 AND 16

Form C						Form D		
Number	a/a_1	u/a_1	Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1	0	14	0.951	0.385	1	1	0
2	1.144	0	15	.970	.150	2	1.144	0
3	1.072	.300	16	.954	.330	3	1.072	.36
4	1.102	.210	17	1.018	.080	4	1.080	.32
5	1.088	.440	18	1.049	.245	5	1	.365
6	1	.880	19	1	.180	6	1.008	-.04
7	1.030	-.150	20	1.001	.355			
8	1.060	0	21	1	.490			
9	.912	.440	22	.969	.575			
10	1	.350	23	.902	.360			
11	.954	.230	24	.911	.245			
12	.950	.500	25	1.012	.060			
13	.984	.390	26	1	.018			

TABLE IV - STATE VALUES FOR FIGURE 17

Form E

Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1	0	27	1.028	-0.017
2	1.144	0	28	1.028	-.035
3	1.133	.200	29	1.017	-.075
4	1.039	.440	30	1.007	-.025
5	1.109	.245	31	[illegible]	.475
6	1.100	.510	32	.980	.050
7a	1.013	.013	33	.990	.110
7b	1	1	34	1	0
8	1.002	0	35	1	.100
9	1.042	.100	36	1.070	.100
10	1.040	.200	37	1	.110
11	1.080	0	38	.969	.380
12	1.055	.125	39	.924	.570
13	1.050	.250	40	.950	.250
14	.955	.725	41	.926	.310
15a	1.013	.455	42	.970	.150
15b	1	.51	43	.978	.190
16	1.012	0	44	.966	.190
17	1.049	.087	45	1.016	.040
18	1.015	.075	46	1.013	.010
19	1.055	-.125	47	1.013	.010
20	1.030	0	48	1.015	.035
21	1.020	.050	49	1.013	.080
22	1.020	.100	50	.990	.160
23	.925	.573	51	1.042	-.2
24	.970	.360			
25	1	.210			
26	1.042	-.100			

TABLE V - STATE VALUES FOR FIGURE 18

Form F

Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1.000	0	12	0.4[illegible]	-0.03
2	1.144	0	13	.956	.20
3	1.072	.36	14	.986	-.01
4	1.03	.57	15	.914	.41
5	1.064	.27	16	1.070	.35
6	1.000	.54	17	.946	.77
7	1.000	.54	18	1.010	.35
8	.946	.27	19	.916	.42
9	.936	.21	20	[illegible]	.39
10	.916	.42	21	[illegible]	-.05
11	.910	.13			

TABLE VIII - STATE VALUES FOR FIGURE 21

Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1.141	0	22	[illegible]	-0.20
2	1.072	.36	23	1.074	.02
3	1.094	.27	24	.914	.23
4	1.084	.40	25	.944	.40
5	1.110	.29	26	1.0[illegible]	.06
6	1.053	.13	27	[illegible]	.11
7	1.123	.37	28	[illegible]	.05
8	1.110	[illegible]	29	[illegible]	.09
9	1.030	1.10	30	[illegible]	.07
10	.990	.50	31	[illegible]	.14
11	1.032	-.16	32	[illegible]	.07
12	1.073	-.13	33	.914	.02
13	1.022	-.17	34	[illegible]	.74
14	1.022	.11	35	[illegible]	.17
15	1.026	.02	36	[illegible]	.08
16	1.026	.15	37	.978	.16
17	1.060	.15	38	1.007	.24
18	1.023	-.15	39	.970	.20
19	1.004	[illegible]	40	.950	.04
20	1.020	-.13	41	.981	.06
21	.994	-.03	42	1.144	0

TABLE VII — STATE VALUES FOR FIGURE 20

Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1.000	0	29	0.998	0.02
2	1.144	0	30	1.026	-.13
3	1.072	.36	31	1.010	-.05
4	1.108	.18	32	.980	-.07
5	1.000	0	33	.982	.09
6	1.036	.18	34	1.008	-.04
7	1.054	.25	35	1.008	-.03
8	1.064	.42	36	.998	-.01
9	1.110	.20	37	.984	.05
10	1.058	.48	38	.986	.07
11	1.102	.45	39	.958	.09
12	1.058	.07	40	.938	.06
13	1.044	.14	41	1.030	.07
14	1.044	.22	42	1.022	.11
15	1.070	.09	43	1.042	.01
16	1.051	.17	44	1.050	.07
17	1.051	.27	45	1.022	.11
18	1.003	.36	46	1.026	.04
19	1.022	1.072	47	1.028	-.06
20	1.000	.29	48	1.010	0
21	1.022	-.11	49	1.008	.04
22	1.008	-.04	50	1.020	-.10
23	1.020	-.14	51	1.008	-.04
24	1.020	-.16	52	1.028	.06
25	1.004	-.08	53	1.020	.10
26	.994	-.03	54	1.008	.04
27	.980	.04	55	.988	.03
28	.980	.11			

TABLE IX - STATE VALUES

For Figure 25			For Figure 26		
Number	a/a_1	u/a_1	Number	a/a_1	u/a_1
1	1	0	1	1	0
2	1.144	0	2	1.144	0
3	1.108	.16	3	1.108	.16
4	1.072	0	4	1.072	0
5	1.072	.36	5	1.072	.36
6	1.036	.16	6	1.036	.18
7	1.01	[illegible]	7	1.004	.12
8	1.036	[illegible]	8	1.076	.14
9	1	.72	9	1	.72
10	1	.56	10	1	.36
11	.964	.54	11	.964	.54
12	1	.56	12	1	.36
13	.974	.23	13	.998	.30
14	.938	.41	14	.962	.48
15	.974	.27	15	.919	.30
16	.964	.18	16	.964	.18
17	.928	.78	17	.928	.36
18	.904	.24	18	.904	.24
19	.966	-.07	19	.952	0
20	.990	.05	20	.976	.12
21	1	.1	21	1	.24

TABLE VI - STATE VALUES FOR FIGURE 19

Number	a/a_1	u/a_1
1	1	0
2	1.144	0
3	1.072	.36
4	1.094	.25
5	1.094	.420
6	1.11	.29
7	1.096	.43
8	1.022	1.022
9	1.023	-.13
10	1.022	-.11
11	1.044	0
12	1.022	.11
13	1.02	-.1
14	1.02	-.16

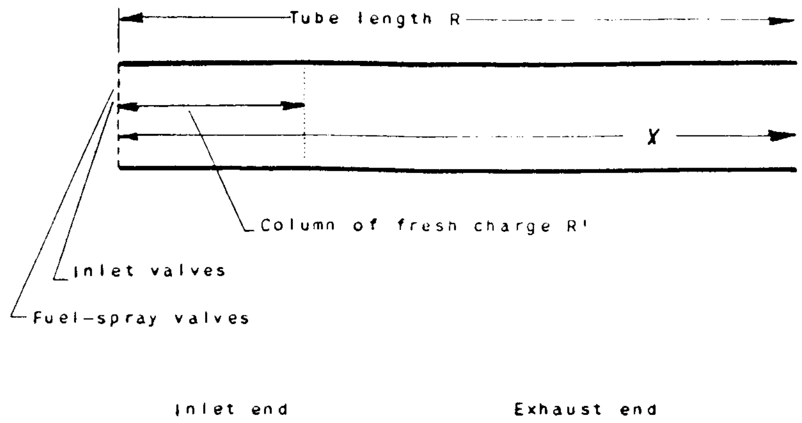


Figure 1. - Schematic diagram of a jet tube.

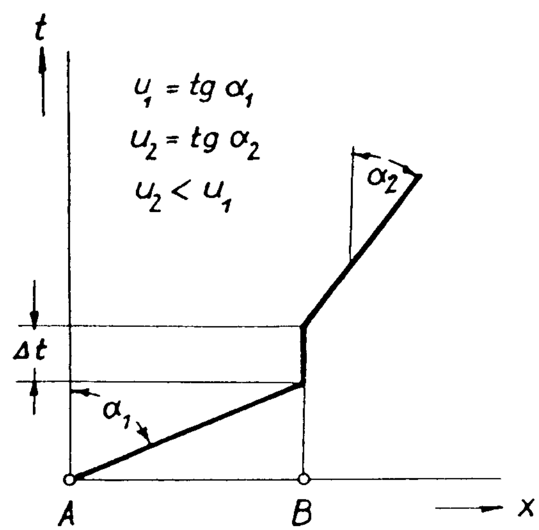


Figure 2. - Graphic representation of a timetable in the t,x diagram.

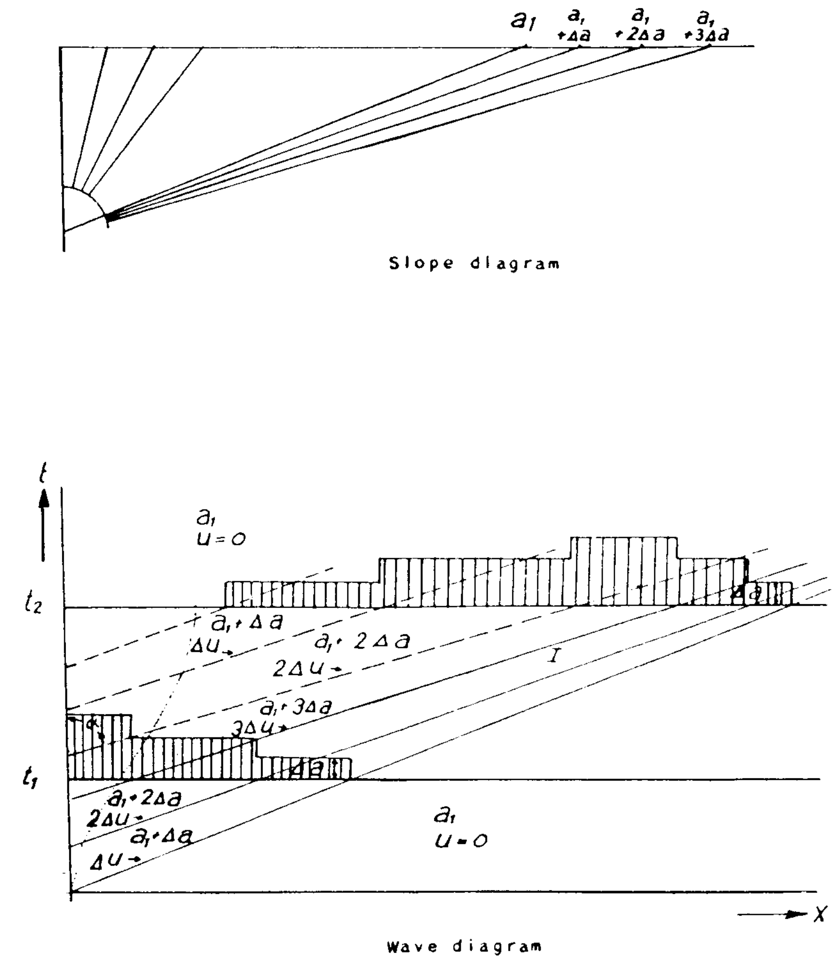


Figure 3. - Wave propagation in t,x plane. The form at two moments t_1 and t_2 of wave approximately represented by individual wave elements.

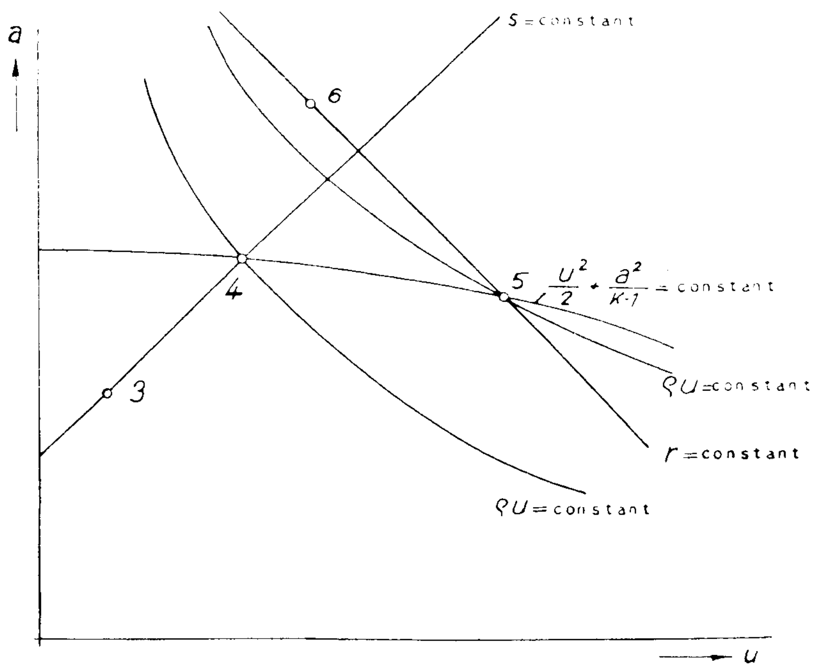


Figure 4. – Passage of a wave through change in cross section as represented in a,u diagram.

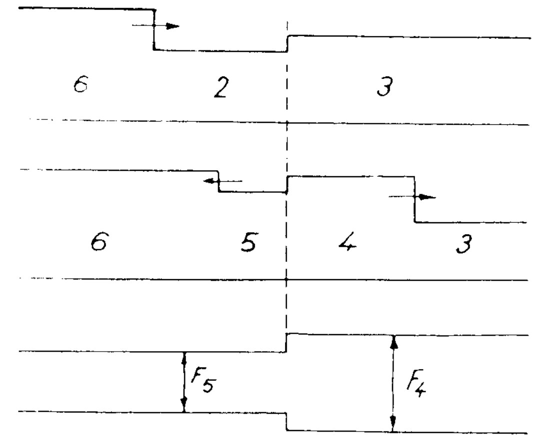


Figure 5. – Passage of a wave through change in cross section. Momentary states.

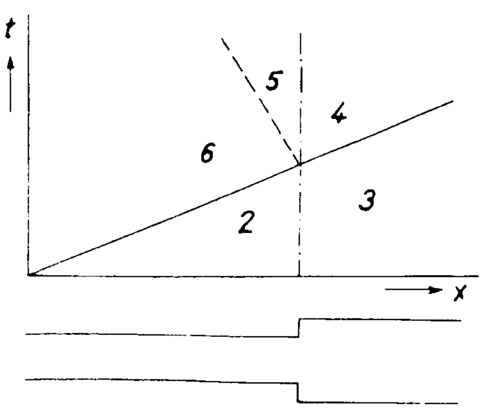


Figure 6. – Passage of a wave through change in cross section as represented in t,x diagram.

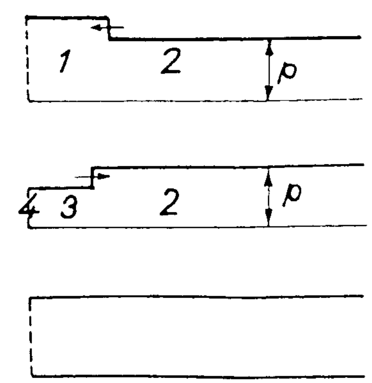


Figure 7. – Reflection of rarefraction wave at inlet valves. Momentary states before and after reflection.

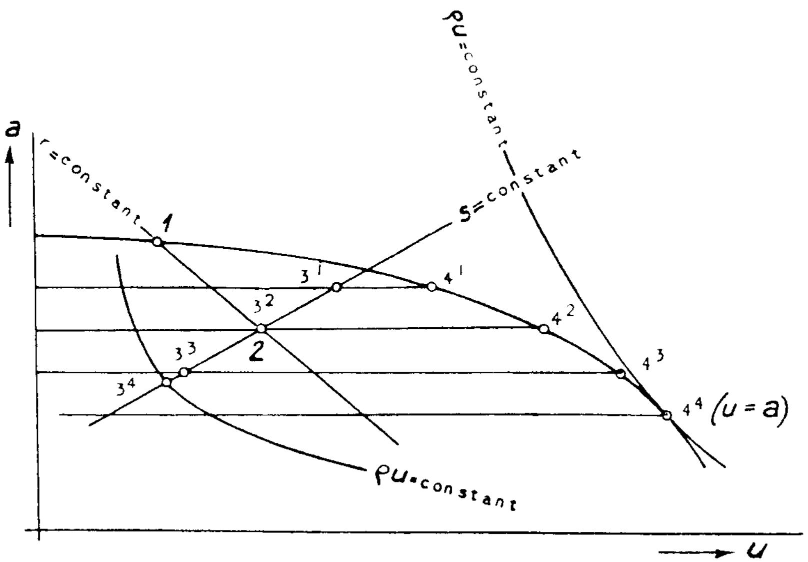


Figure 8. - Reflection of a rarefaction wave at inlet valves. Determination of reflected wave in a,u diagram.

Four cases:

Large valve cross section, reflection of condensation wave 3^1

Smaller valve cross section, no reflection because $3^2 = 2$

Still smaller valve cross section, reflection of rarefaction wave 3^3

Still smaller valve cross section, reflection of rarefaction wave 3^4 and flow through valves at velocity of sound

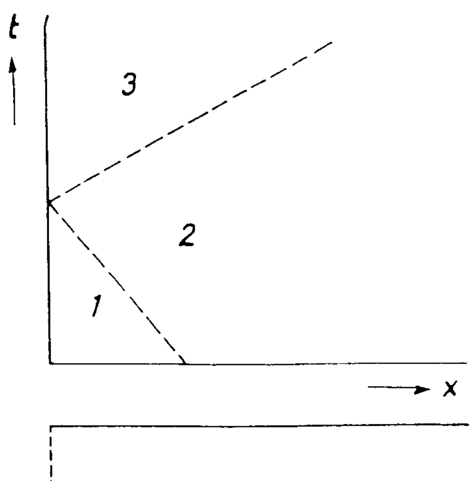


Figure 9. - Reflection expressed in t,x diagram. (To accompany figs. 7 and 8.)

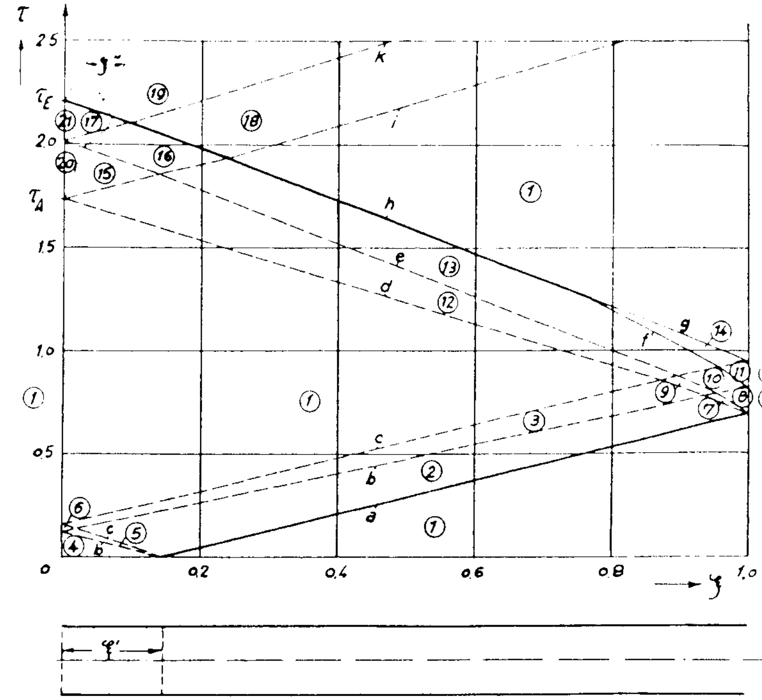


Figure 10. - Wave propagation in jet tube. $F_E/F_R = 0.2$.

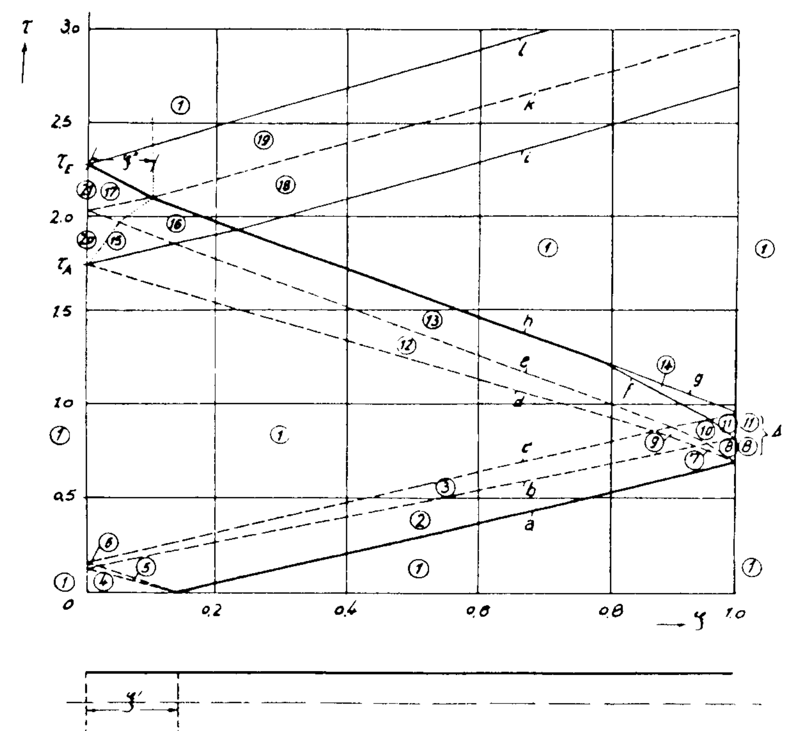


Figure 11. - Wave propagation in jet tube. $F_E/F_R = 0.4$.

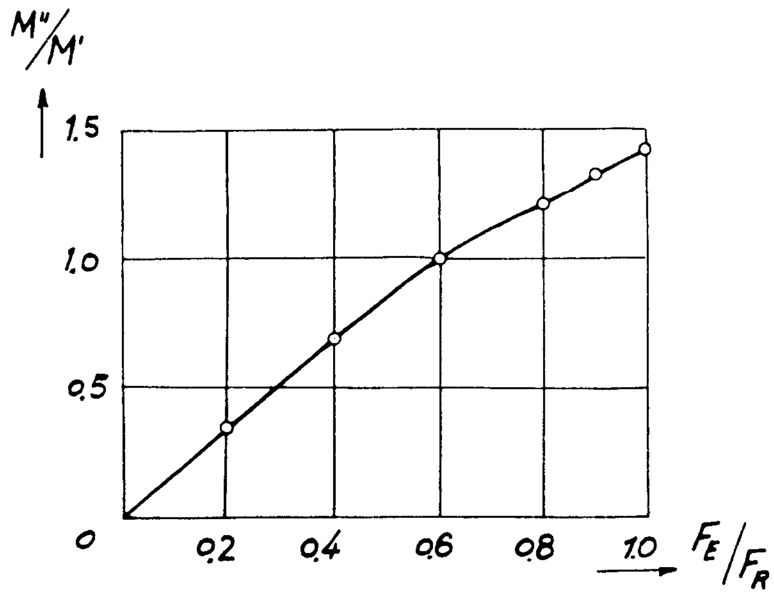


Figure 12. - Ratio of quantity of newly induced charge to quantity originally present as function of F_E/F_R in stationary jet tube.

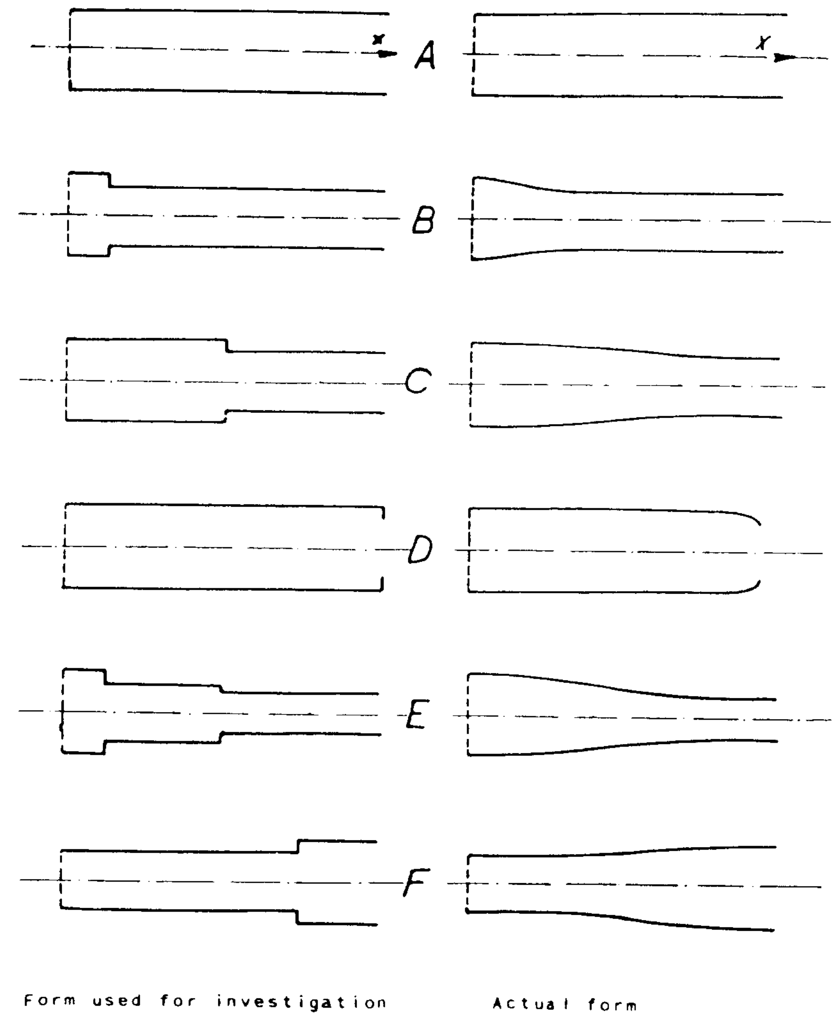


Figure 13. - Tube forms investigated. From left to right each smaller cross section is 50 percent of the preceding one.

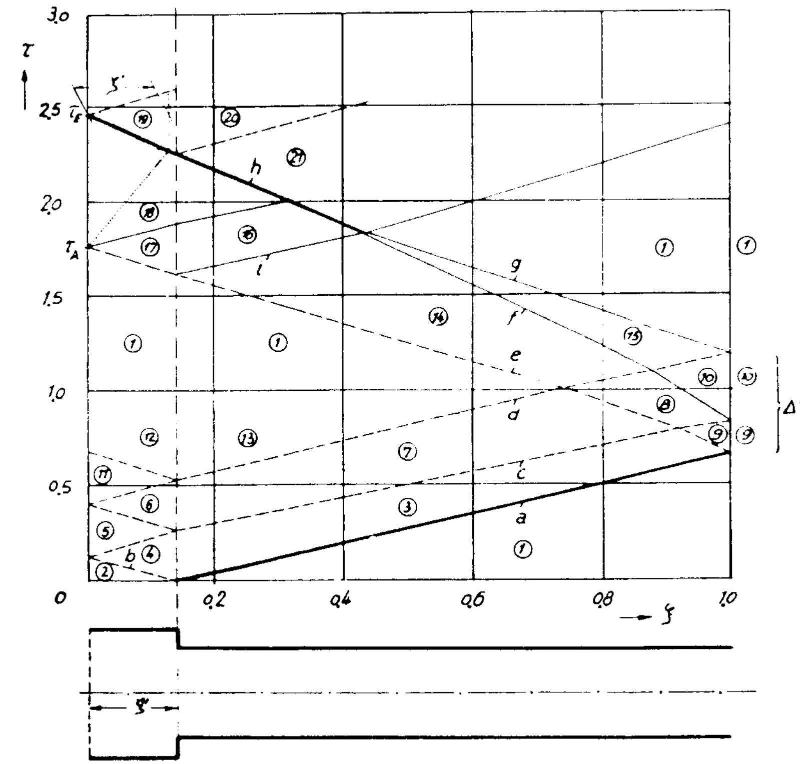


Figure 14. - Wave propagation in the jet tube. $F_E/F_R = 0.4$; form B.

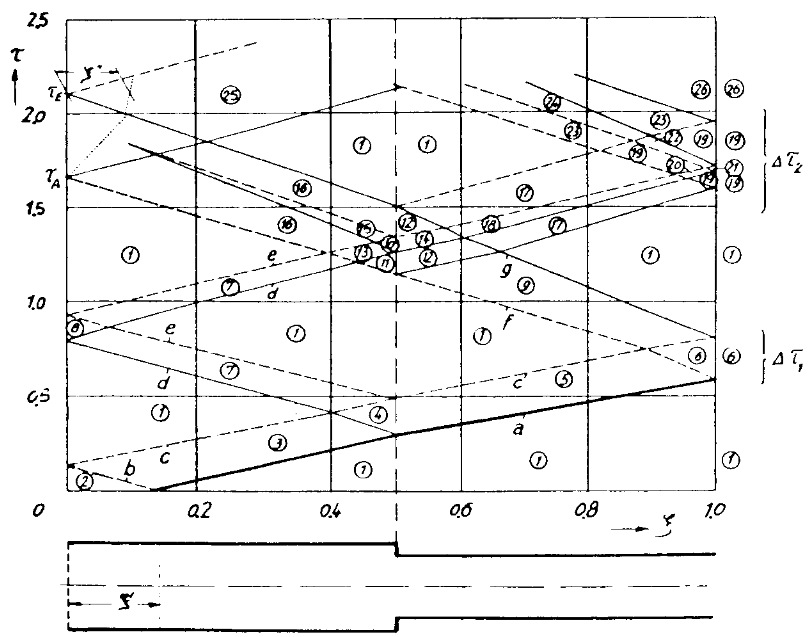


Figure 15. - Wave propagation in jet tube. $F_E/F_R = 0.4$; form C.

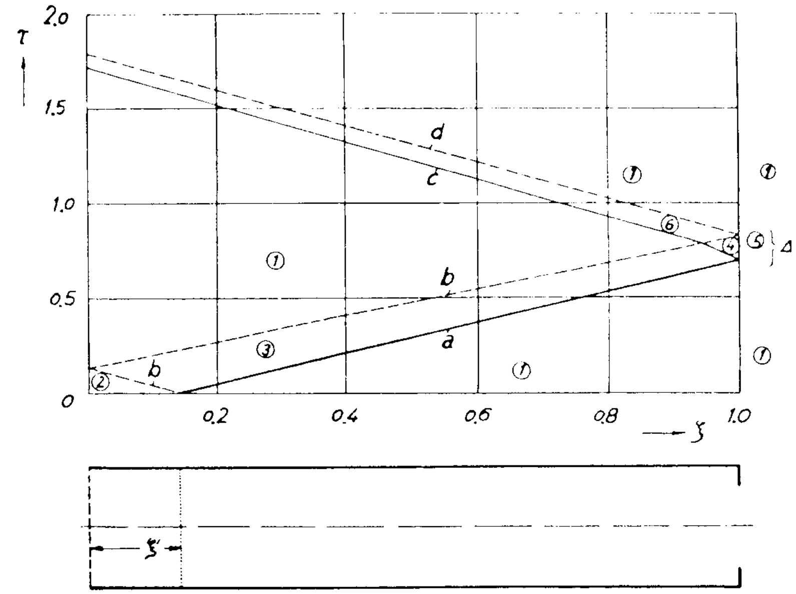


Figure 16. - Wave propagation in the jet tube. $F_E/F_R = 0.4$; form D.

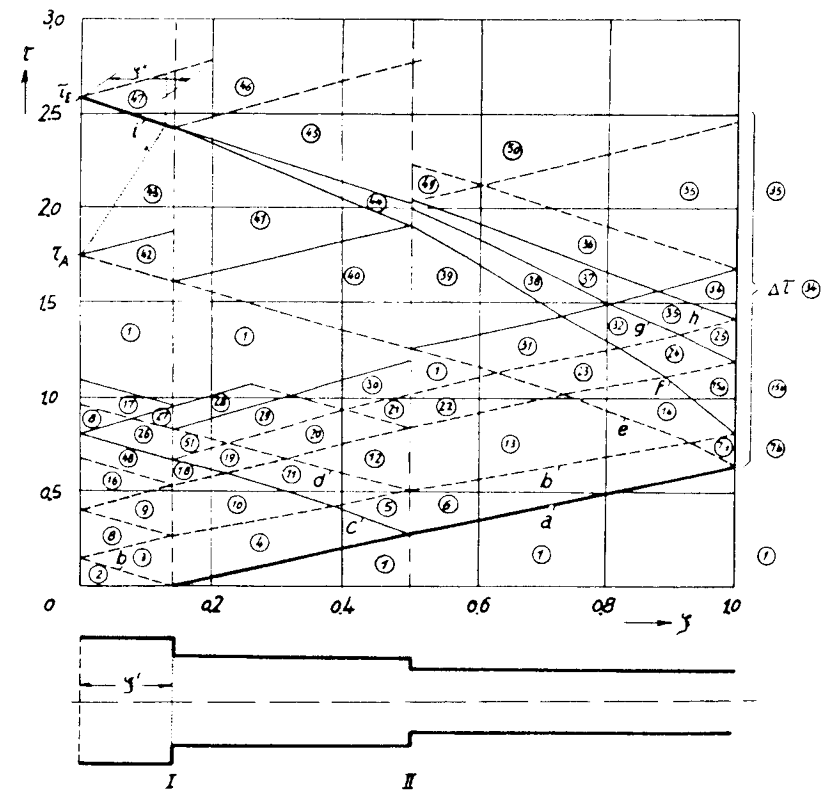


Figure 17. - Wave propagation in the jet tube. $F_E/F_R = 0.4$; form E.

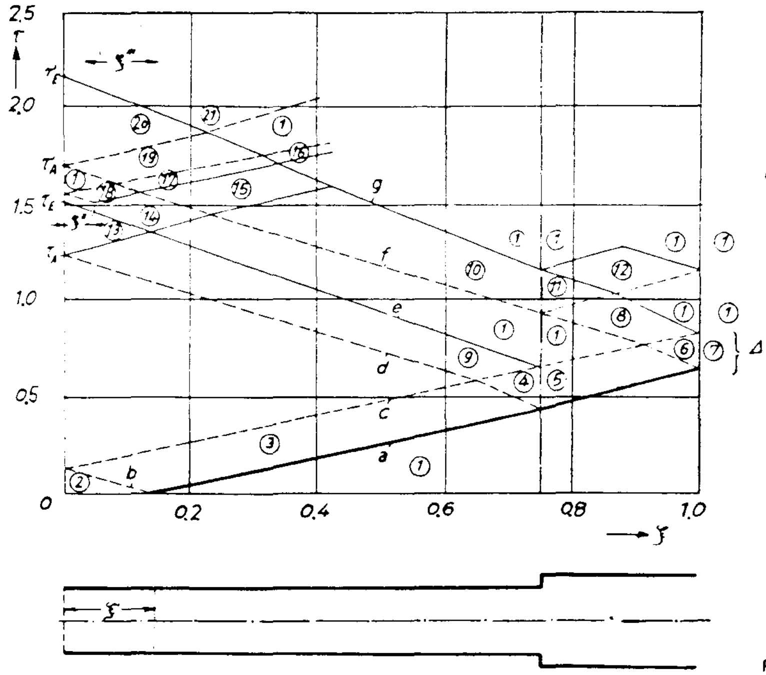


Figure 18. - Wave propagation in the jet tube. $F_E/F_R = 0.4$; form F.

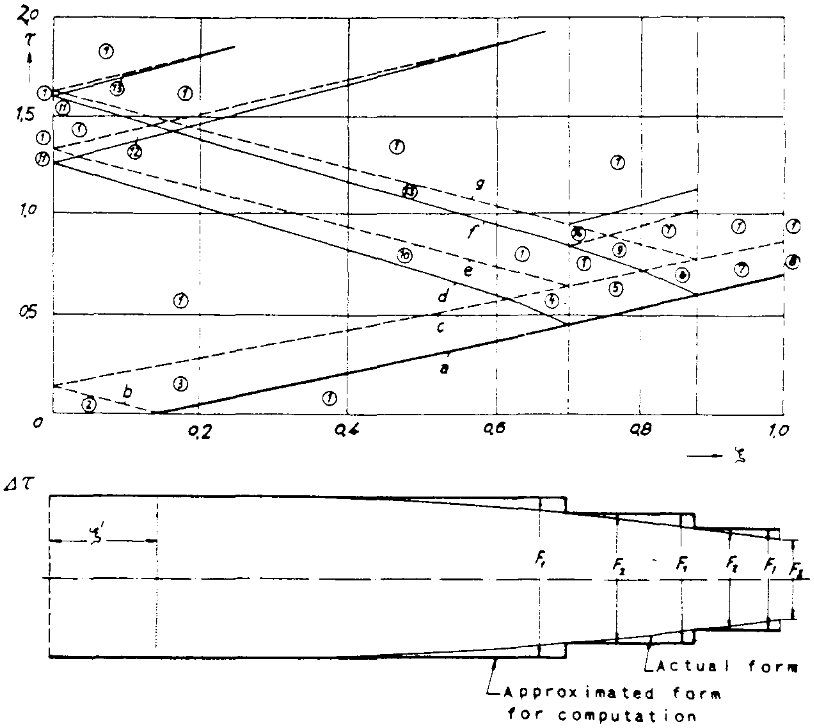


Figure 19. - Wave propagation in the Argus VSR9z jet tube. $F_E/F_R = 0.4$.

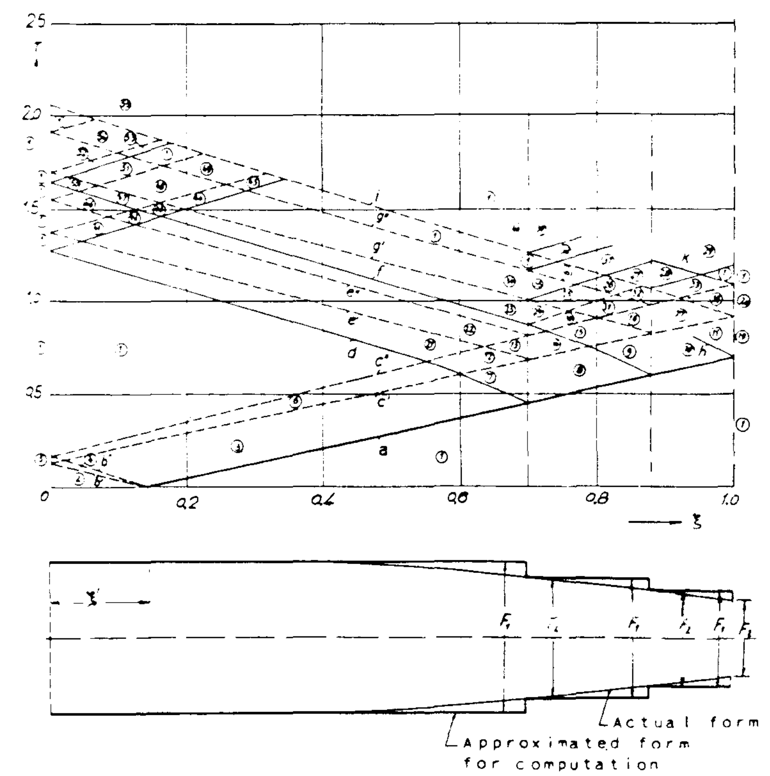


Figure 20. - Wave propagation in the Argus VSR9z jet tube. $F_E/F_R = 0.4$. (Closer approximation than in fig. 19.)

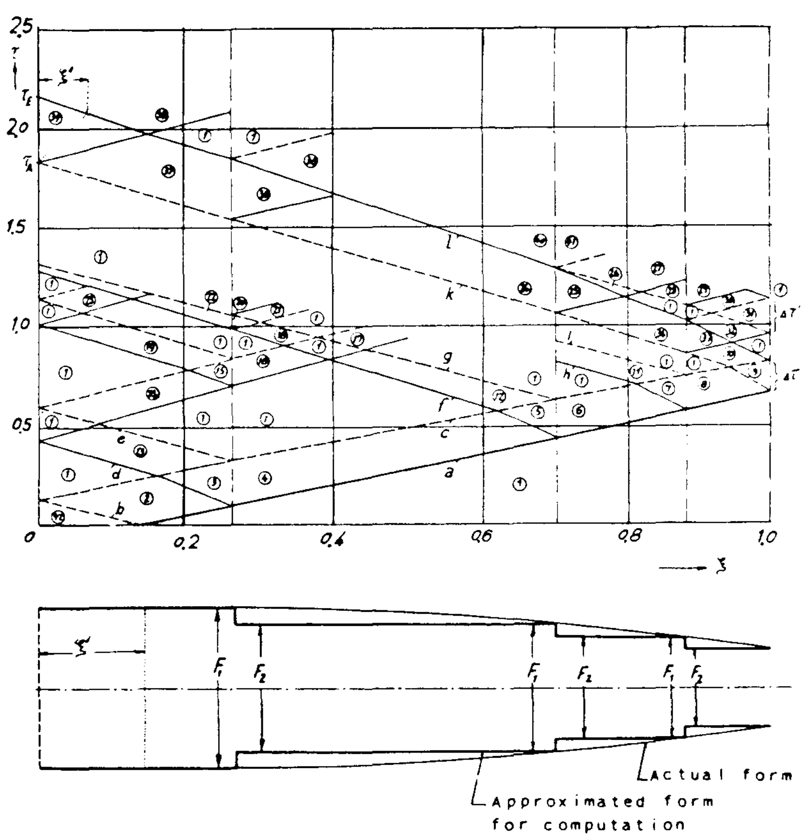
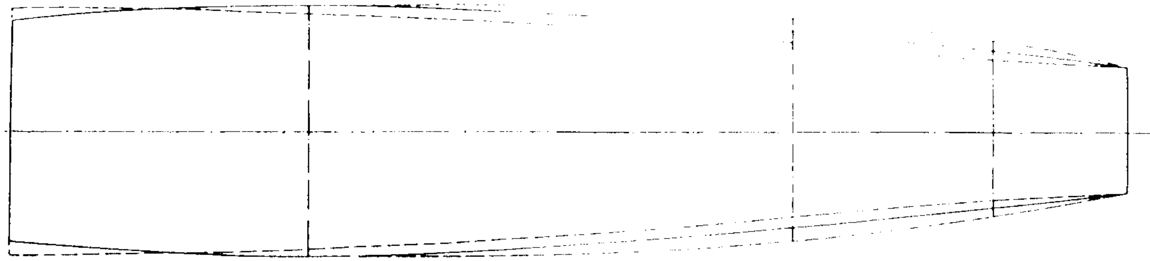


Figure 21. - Wave propagation in the Argus VSR9z jet tube. $F_E/F_R = 0.4$.



-------- Mean curves for two step-curves as approximations

———— Actual curves

Figure 22. - Longitudinal section through VSR92.

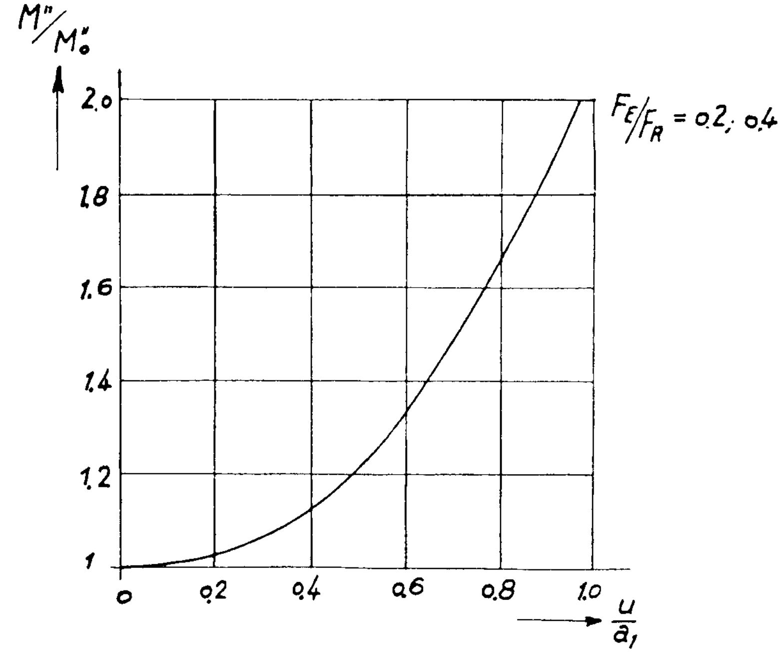


Figure 24. - Ratio of newly indrawn quantity of fresh charge M" to indrawn quantity M''_o when u = 0 as function of flight speed u.

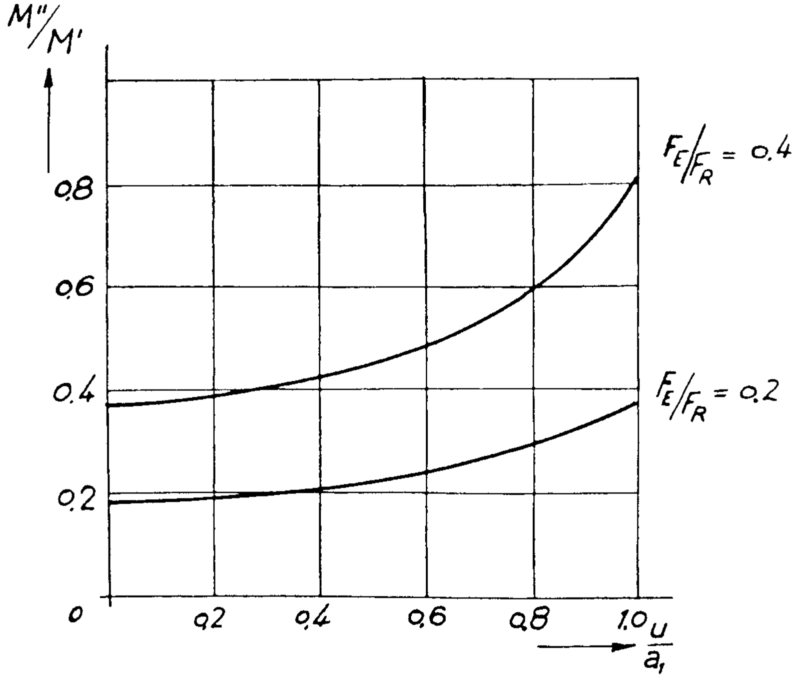


Figure 23. - Ratio of newly indrawn quantity of fresh charge M" to quantity originally present M' as function of flight speed u.

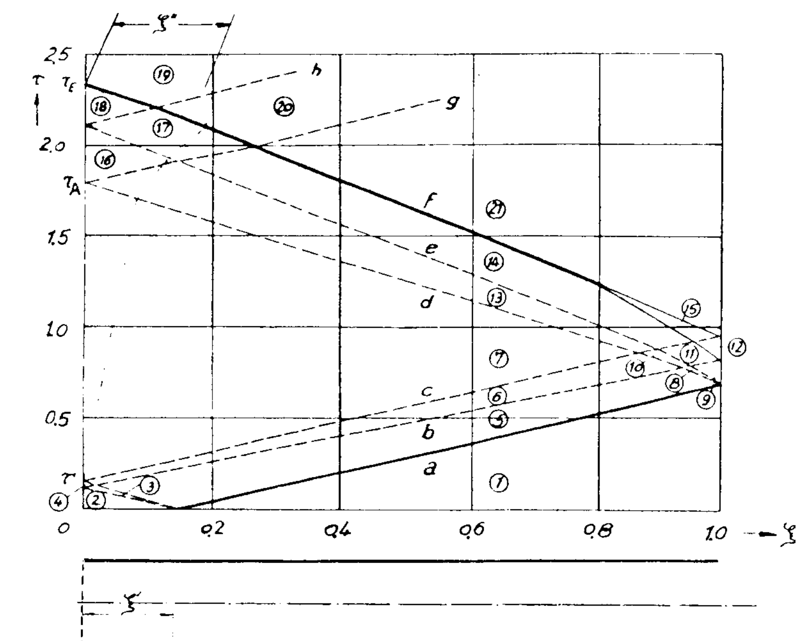


Figure 25. - Wave propagation in jet tube in the case of leakiness in the valves. F_E/F_R = 0.2; leakiness 50 percent, that is $\Delta F_E/F_R$ = 0.1; flight speed = 0.64 of velocity of sound.

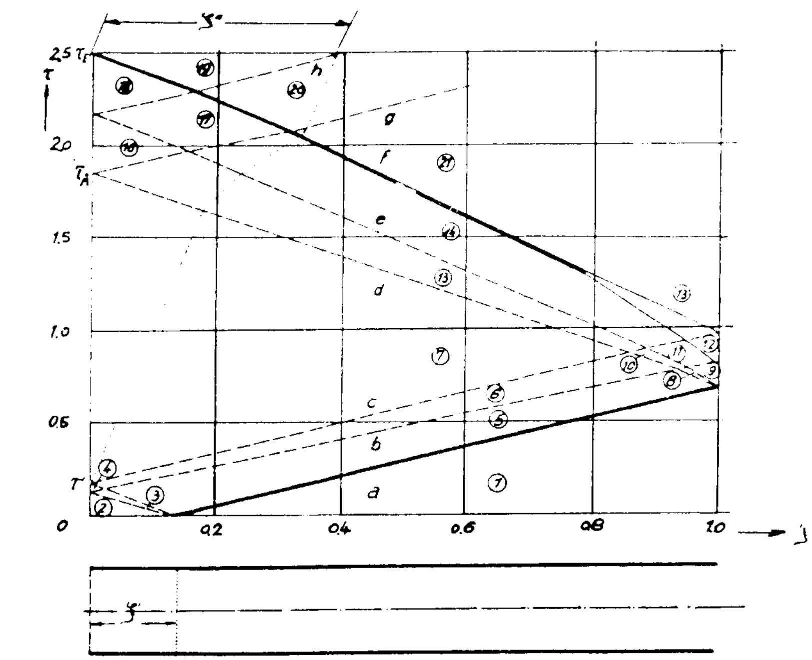


Figure 26. - Wave propagation in jet tube in the case of leakiness in the valves. F_E/F_R = 0.2; leakiness, 100 percent, that is $\Delta F_E/F_R$ = 0.2; flight speed = 0.64 of velocity of sound.

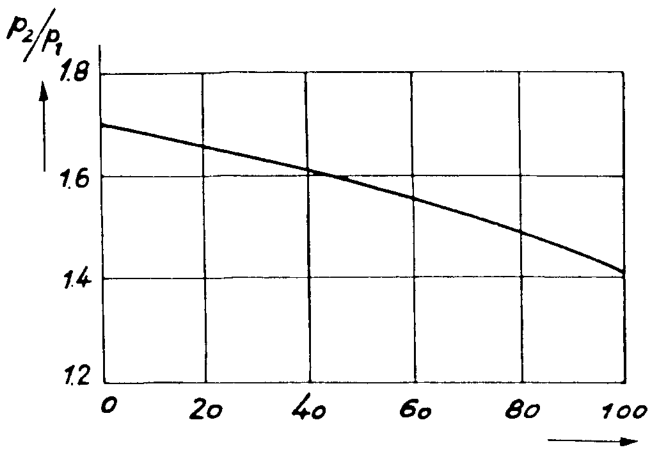


Figure 28. - Pressure ratio in returning (igniting) compression shock f plotted against leakiness. (Same conditions as in fig. 27.)

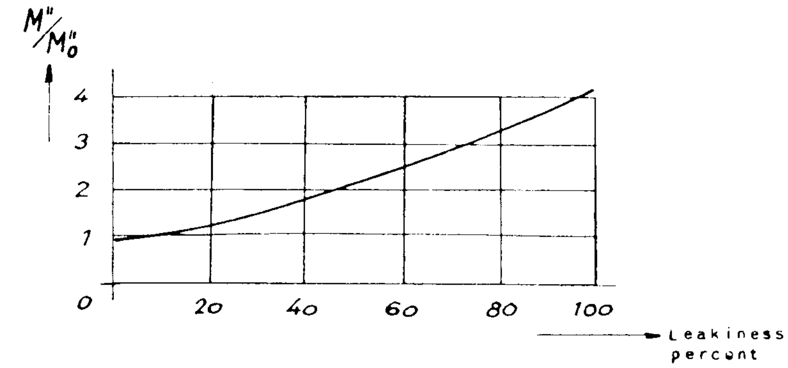
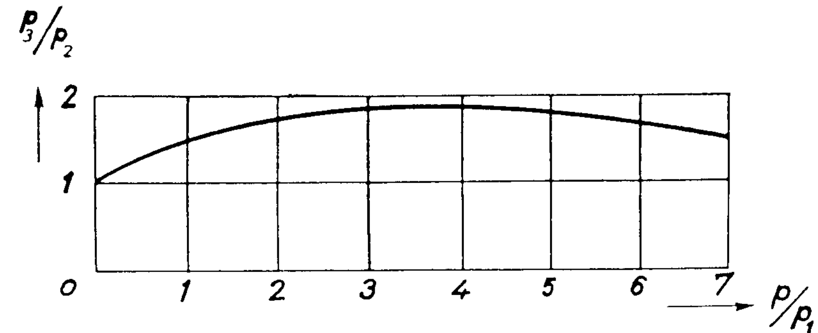
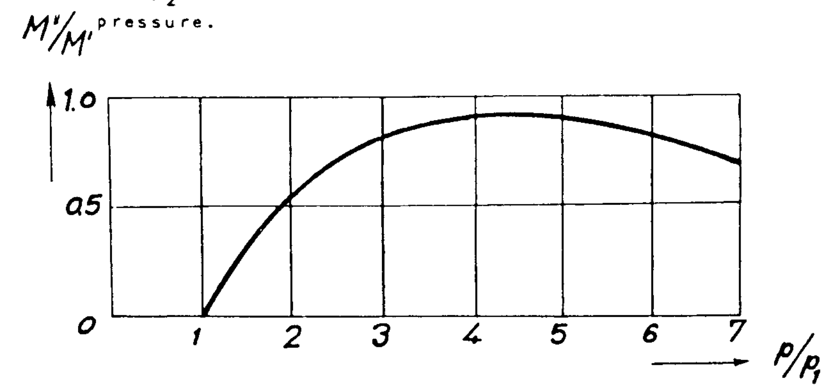


Figure 27. - Ratio of quantity of newly indrawn gas M" to quantity drawn in with zero leakiness M''_0 plotted against leakiness in percentage of fully opened valve cross section F_E/F_R = 0.2. Flight speed = 0.64 of velocity of sound.

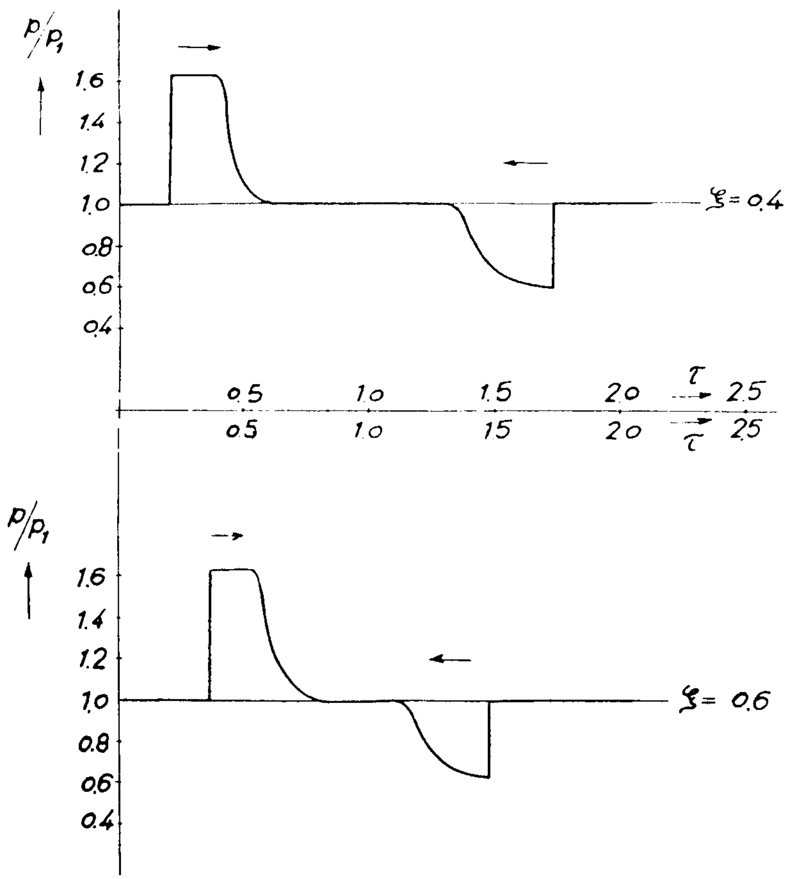


Pressure ratio p_3/p_2 in reflected shock 3 (p_3 before and p_2 after the shock) plotted against combustion pressure.



Ratio of newly indrawn quantity of fresh charge M" to quantity originally present M' plotted against combustion pressure.

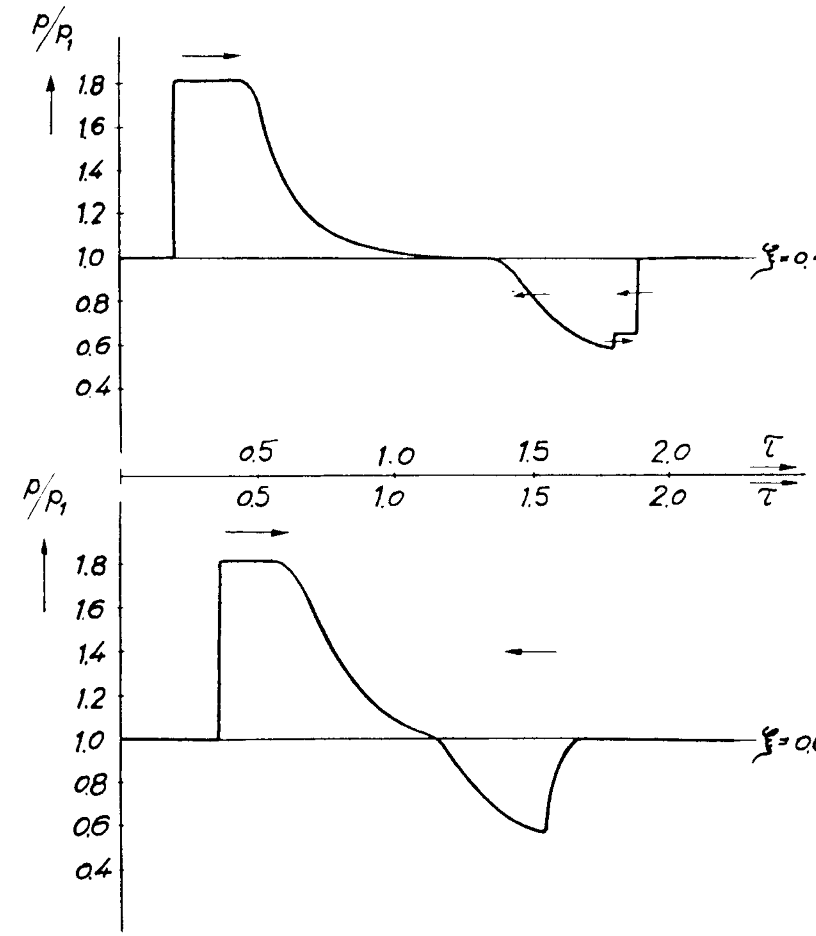
Figure 29. - Influence of combustion pressure p.



→ Outward-traveling wave (+ξ direction)

← Inward-traveling wave (−ξ direction)

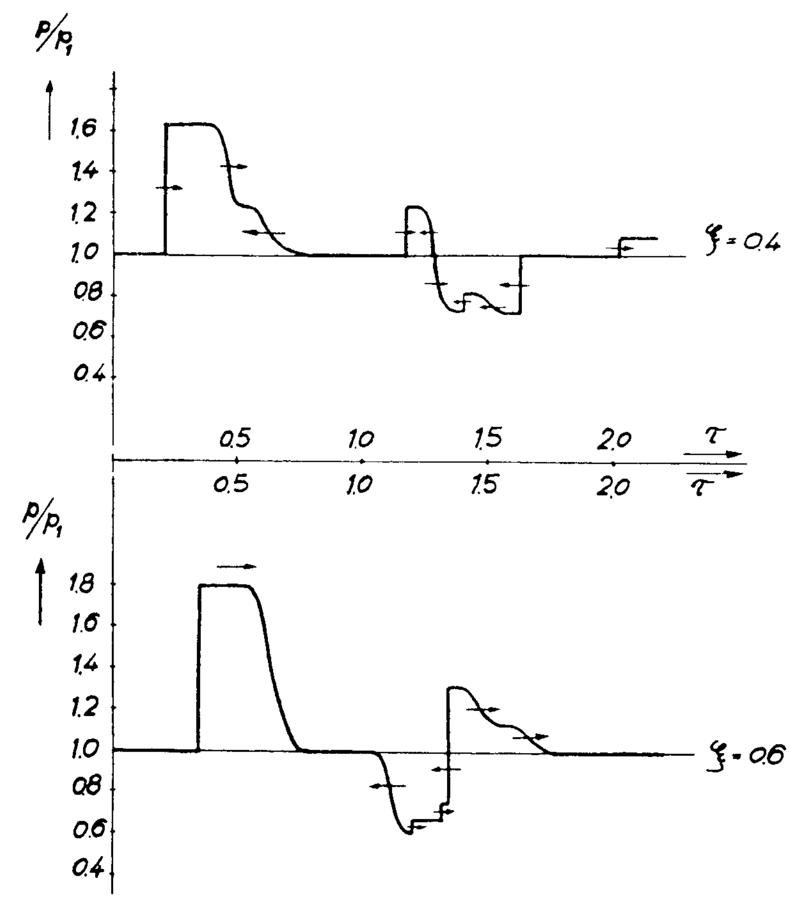
Figure 30. - Variation of pressure with time at cross sections. $\xi = 0.4$; $\xi = 0.6$; form A (cylindrical tube). (See fig. 11.)



→ Outward-traveling wave (+ξ direction)

← Inward-traveling wave (−ξ direction)

Figure 31. - Variation of pressure with time at cross sections. $\xi = 0.4$; $\xi = 0.6$; form B. (See fig. 13.)



→ Outward-traveling wave (+ξ direction)
← Inward-traveling wave (−ξ direction)

Figure 32. - Variation of pressure with time at cross sections. $\xi = 0.4$; $\xi = 0.6$; form C (See fig. 13.)

GAS-DYNAMIC INVESTIGATIONS OF THE PULSE-JET TUBE

PART II

SUMMARY

This investigation will account for the important practical observation made by Paul Schmidt that the ratio of the effective valve cross-sectional area to the tube cross section may not be of any random magnitude and will explain why at too great flight speeds the jet tube ceases to operate. Chemical and thermodynamic processes (for example, constituents or mode of fuel-air-mixture formation or heat losses) are unimportant in this regard.

INTRODUCTION

In Part I the jet tube was investigated using the simplest possible assumption as to the initial condition, namely, that a compressed column of fresh charge suddenly expands. The simplification thus obtained permitted an elementary insight into the nonuniform gas motion, even in noncylindrical tubes. The investigation attributed the setting-off of the explosion to a compression shock wave reflected from the end of the tube and it was useful in that it enabled a prediction as to which tube forms that were aerodynamically desirable would also be capable of operating. It was shown that in tubes tapering toward the rear, the end portion of the tube must be cylindrical.

The investigations of Beckert and Sauer are based on very similar simplified initial conditions.

But the combustion does not take place suddenly; instead, it extends over a rather large part of the working cycle. The consequent variation of pressure with time will be investigated in the cylindrical tube. Because nothing is known of the combustion as a function of the condition values, an experimentally determined value shall be used for the increase of pressure due to combustion; and the simplified assumption shall be made that the change of state takes place adiabatically and simultaneously along the whole column of fresh charge. This assumption presupposes combustion taking place in that manner. In actual fact, the combustion takes place not solely from the end cross section of the column of fresh charge but from the outside inwardly along its whole length due to the continuous presence of burning remnants of the previous charge.
Accordingly for the purposes of this investigation, the only effect of the fuel is the production of a change of state of the fresh charge. The influence of its mass upon the motion will be neglected.

The exhaust gases are assumed to have the same specific heats as the fresh charge and their state is assumed to lie on the same adiabatic curve as that of the fresh charge.

These assumptions imply that in the jet tube a homogeneous gas initially exists throughout under uniform pressure, upon which, confined in a limited space, an adiabatic change of condition is externally imposed.

Comparison with Paul Schmidt's experimental results will show whether a gas flow corresponding to reality arises under the premises thus assumed.

This investigation will be carried out using the Riemann theory of nonuniform gas flow, according to which the gas flow is determined by the propagation of pressure waves. The propagation will be graphically represented in a time-distance diagram. The method of approximation (reference 1) to be used for this purpose has been described in Part I and here is assumed to be familiar.

GAS FLOW DURING AN EXTERNALLY CAUSED CHANGE OF PRESSURE

With an increase of pressure in the column of fresh charge there arises a pressure difference as compared to the exhaust gases adjoining the rear face of this column. This pressure difference results in a movement of expansion, which is transmitted to the exhaust-gas column as condensation waves and to the fresh-charge column as rarefaction waves, in the manner indicated in the time-distance diagram of figure 1'. The assumptions that the change of state is isentropic and that the entropies of fresh charge and exhaust gases are the same lead to the further assumption that these waves are produced in pairs consisting of a condensation wave and a rarefaction wave of equal strength.

With this method of approximation, only a few of the continuous series of waves that are transmitted shall be followed; also as an approximation, constant gas and sound velocities between them shall be assumed. Likewise, the continuous externally imposed increase in pressure will be approximated by separate small jumps, the position and magnitude of which will be determined by the experimentally observed sequence of pressure variations. It is assumed that these abrupt changes take place simultaneously throughout the whole column of fresh charge. They are represented in figure 1' by solid horizontal lines.

With the definition of the manner which the externally imposed pressure varies with time, the simplest premises possible for its variation with the distance x shall be defined. Let the imposed pressure change be taken as uniform throughout the column of fresh charge. It is then obvious that this change will have no direct effect on the motion of the gas but only a later indirect effect caused by the successively generated waves that are transmitted from the end of the column of fresh charge. Pressure variations corresponding to the already existing waves remain; the waves do not change in strength but do change their rate of propagation because of the change in the velocity of sound.

In order to conduct the investigation, the relation is required between the change in gas velocity Δu and the change in velocity of sound Δa resulting from the wave

$$\Delta u = \pm \frac{2}{\kappa - 1} \Delta a$$

and the expression for the velocity of propagation of a wave

$$w = \pm a + u$$

The upper sign applies to a wave traveling in the +x direction; the lower, to one traveling in the -x direction.

COMPUTATION OF GAS MOVEMENT DUE TO GIVEN RISE IN PRESSURE IN COLUMN OF FRESH CHARGE

From pressure measurements on jet tubes, it is found that the pressure rise during combustion corresponds, for the purposes of this investigation, to an increase in the sonic velocity $\Delta a = 0.04 a_1$ in 1/15 the time required for a wave to traverse the jet tube with a sonic velocity a_1 of the atmosphere at rest. On this basis, the rise to the normal value of 2.5 times atmospheric pressure requires an interval of about 1/5 of the working cycle.

According to the observations of Paul Schmidt, the column of fresh charge extends for 1/7 the tube length. At the beginning of the first working cycle, the fresh-charge column is allowed to expand. Let the quantity of charge distributed through 1/7 of the tube length be the proper quantity for continuous tube operation, the quantity that must be drawn in during each period. This fixes the maximum pressure during a cycle because that pressure substantially determines the quantity drawn in for the next period. Thus for the first cycle a maximum pressure of 5.5 times atmospheric and for the second cycle 3.5 times is obtained. The reason for the lower value for the second period is that a motion process already exists from the first period.

The ratio of maximum effective valve area to tube area is set at 0.4. The reflection of the waves at the valves has been explained in Part I. A rough allowance shall be made, however, for the mass and stiffness of the springs by assuming a gradual opening of the valves to their maximum stroke.

The initial phases of tube operation will be treated here rather than the ultimate oscillatory action because the Riemann theory refers to the initial phases. In view of the rather burdensome work involved, only the first two periods will be developed. The second period will give a sufficient idea of the essentials of the ultimate oscillatory action.

The wave propagation in the time-distance diagram has been constructed in figures 7' to 11'. Here the condition of motion and state of the gas at all tube cross sections and at all times may be found and along x = constant and t = constant lines the time sequence of pressure at a given cross section and the distribution of pressure along the axis of the tube at a given time may be read. As coordinates, take dimensionless time $\tau = t \frac{a_1}{l}$ and dimensionless distance $\xi = x/l$, in which l = tube length. The slope of a line of propagation measured from the τ-axis is thus a velocity made dimensionless by dividing it by a_1, the sonic velocity of the initial state of rest. The values of the states existing at each point are shown in figures 7' to 11'. [NACA comment: The pair of numbers shown in each area of figures 7' to 11' are the values of a/a_1 (upper) and u/a_1 (lower), respectively. The pressure may be calculated from the relation $p/p_1 = (a/a_1)^{2\kappa/(\kappa-1)}$.]

In order to permit a better over-all view, figure 2' gives a summary of the wave propagation that is constructed in detail in figures 7' to 11'. In figure 2', pressures above and below atmospheric are indicated by + and - signs, and the direction of the flow by arrows.

The condensation waves A' and A" (fig. 2'), which arise from the combustion in the first and second cycles, initiate an outflow from the end of the tube at greater than atmospheric pressure and at the velocity of sound. Consequently, the subsequent rarefaction waves B' and B" are reflected as rarefaction waves C' and C" until they have liquidated the excess pressure. The still later rarefaction waves D' and D" are reflected as condensation waves E' and E"; they produce a condition of atmospheric rest. This condition exists in the regions designated I.

At the inlet end, the waves C' and C" initiate the intake period. When the first wave arrives, the valves are still closed. From that moment on, due to their inertia, they gradually open. The first waves are consequently reflected as the rarefaction waves F' and F"; the last waves are reflected as the condensation waves G' in the first cycle and not reflected at all in the second.

The first cycle influences the second cycle through waves E', F', and G'. The fresh charge drawn in during the first cycle is compressed and, as indicated in Part I, is also ignited by waves E',

because obviously no other phenomenon exists that might serve to set off the explosion. The second cycle begins with this ignition.

Wave G' combines with the pressure waves originating in the second explosion, thereby reinforcing them.

The rarefaction waves F' are reflected at the end of the tube as condensation waves H' and produce at that point inward flow with a maximum gas velocity of $u/a_1 = 0.16$. The boundary of the inflowing air is shown by a finely dotted line. This air eventually occupies 1/8 of the tube. This phenomenon of inward flow at the exhaust end of the tube was experimentally observed by Paul Schmidt and termed by him "intrusion of air." This phenomenon was considered unimportant and hence not treated in Part I of this report because its occurrence was not so obvious on the basis of the simpler initial conditions assumed in Part I.

The effect of waves H' is to produce, following the combustion, a pressure higher than atmospheric at the inlet end, which exists until the arrival of rarefaction waves C". These waves C" initiate the second intake period.

Figure 3' shows the variation of pressure with time at the inlet end, as derived from the time-distance diagram.

COMPARISON WITH EXPERIMENTALLY OBSERVED RESULTS AND CONCLUSIONS

For comparison, figure 4' shows experimental results obtained by Paul Schmidt. It is apparent that the calculated pressure diagram (fig. 3') substantially agrees with his.

It is of especial interest that an explanation can now be made for the shoulder in the curve E in figure 4', which appears more or less markedly in all observations of pressure at the inlet end. It appears also in the pressure diagram that has been calculated (fig. 3') and is in fact produced by the waves H', which were originally reflected from the open valves. It is thus seen to be a phenomenon of valve operation. The stiffer the valves are and the more mass they have the more this shoulder will peak.

On the other hand the comparison shows that the combustion does not begin suddenly with a constant speed of burning, as has been here assumed for the sake of simplicity. The irregularities at the beginning of the experimentally observed pressure rise must be due to vibration of the valve flaps, which will be disregarded here.

In Part I, the occurrence of the condensation waves E was found as a criterion of the tube operability, from which it was evident that tubes with a constriction at the exhaust end are not operable. A second criterion of operability now appears, one that simply expresses the observation of Paul Schmidt that operability is affected by the opening ratio of the valves, that is, the ratio of maximum effective open cross-sectional area of the valves to the cross-sectional area of the tube. In other words, with too great an opening ratio the tube will not operate.

The explanation for this is found in figure 2'. If the opening ratio is too great, scarcely any of the waves reflected at the valves will be rarefaction waves F' but instead predominantly condensation waves G'. The reflected waves H' are then not condensation but rarefaction waves. This means that the fresh charge will flow in at higher pressure and in greater quantity and the pressure level will thus be raised; but on the other hand, when the reflection of wave H' takes place at the open end of the tube an inflow (rebound), or as Paul Schmidt calls it an intrusion of air, will not occur but instead, due to the reversed character of the waves, a premature explosion will be initiated, namely, before the compression shock wave A" starts the exhaust process. The inflow of fresh charge thus has the effect of a weak intermediate explosion, that is, it creates a condensation wave analogous to G' traveling toward the open end of the tube. But the wave corresponding to H' consequently reflected from the end of the tube is now a rarefaction wave that weakens the explosion.

At the correct smaller valve-opening ratio there are, on the contrary as in figure 2', predominantly rarefaction waves F'. By reflection at the end of the tube, they give rise to the condensation wave H', which produces the intrusion of air and strengthens the explosion. Thus it may be seen that the pressure loss in the valves must be at least so great that the indrawn fresh charge has a pressure lower than that of the exhaust gas in the tube.

It might also be said, that when the valve-opening ratio is too great, the combustion creates too little pressure because it must instead produce volume to maintain the flow through the tube initiated by E'. The combustion thus continues the process that took place due to the open valves, namely, displacement of volume. The limiting case of combustion at constant pressure is approached. The tube is not operable in that case because a certain progressive change of pressure is necessary in order that enough fresh charge may be drawn in for the next period.

It is now also seen that the two requirements stated by Paul Schmidt, namely, not too great a valve-opening ratio and intrusion of air, are mutually interdependent; for it has been learned that with too great a valve-opening ratio the process of continuous flow through the tube is too predominant over the oscillatory process. For the development of sufficient pressure during the explosion, a back-flow at the open end, that is, intrusion of air, is necessary.

INFLUENCE OF FLIGHT SPEED ON OPERABILITY

In the case of the jet tube in flight, the level of pressure on the valves is increased by the amount of the dynamic pressure, as compared with the case of the fixed jet tube. At a sufficiently high flight speed, the increase is so great that the pressure of the indrawn fresh charge is not lower than the exhaust-gas pressure in the tube and the same phenomenon appears as in the case of a too great valve-opening ratio. Thus at a certain flight speed, the tube will cease to be operable for the same reasons as in the case of a too great valve-opening ratio.

These operating limits should be of more significance than those computed by Beckert from the limits of combustion set by too lean and too rich fuel mixtures because the combustion limits are quite broad for the range of atmospheric conditions that will be encountered and may easily be influenced by modifications of design.

In addition to this previously described injurious effect of flight speed, there is the effect pointed out in Part I, namely, that with increasing flight speed the compression shock that sets off the explosion also becomes weaker, assuming the valve-opening ratio is kept constant.

SHIFTING OPERATING LIMITS TO PERMIT HIGHER FLIGHT SPEEDS

An extension of the operating limits would seem to be possible through automatic regulation of the valve-opening area in accordance with flight speed because a reduction in the opening ratio with increasing flight speed would fulfill the requirement of a sufficiently great pressure loss in the valves. However, it must also be made sure that a sufficient quantity of fresh charge is drawn in.

This consideration sets the limit to the possibilities of opening-ratio regulation. In this connection, figure 23 of Part I, which is duplicated as figure 5' in this part, shows that with constant opening ratio increasing flight speed has little effect on the indrawn quantity of fresh charge up to a speed of 0.4 of the velocity of sound; after this point the effect is more marked. The regulating mechanisms therefore ought not to be operative until the higher flight speeds are attained. Figure 5' also gives a basis for estimating the possible scope of the regulation. At a flight speed of 0.8 of the velocity of sound, the reduction in the opening ratio ought to be about 25 percent.

Another simpler possibility is suggested in figure 6'. The full effect of the impact pressure is prevented by means of a cap in front of the tube. This cap will also reduce the flow resistance. The cap can be so formed that approximately atmospheric pressure will be attained at the annular slit where the fresh charge enters, regardless of flight speed. The introduction of air through the slit must, of course, be so arranged that turning losses are avoided.

REDUCTION OF INTERNAL-FLOW RESISTANCE

It may be asked whether it is possible to induce the oscillatory process by means other than valves with high flow resistance. One possibility would appear to be the positive mechanical regulation of the inlet area. It would then be possible to choose a relatively greater opening ratio maintained over a shorter time, rather than a smaller opening ratio operative for a relatively long time. On the one hand, the flow losses would then be less; and on the other, the opening of the valves could be made to occur as late as possible and with maximum suddenness. Late opening would produce strong rarefaction waves F' because the valves would still be closed when the first waves C' struck them; and sudden opening would result in strong condensation waves G'. Both these effects would strengthen the subsequent explosion and thereby increase the thrust.

SUMMARY

By taking into account the course of the development of pressure by combustion, a new insight has been obtained into the processes of motion within the jet tube, an insight that explains a number of empirical observations, namely: certain particulars of the sequence of pressure variations; the existence of an optimum valve-opening ratio; the occurrence of an intrusion of air; and the existence of a flight speed above which the jet tube ceases to operate.

At too great an opening ratio or at too great a flight speed, the continuous flow through the tube is too predominant over the oscillatory process to permit the occurrence of an explosion powerful enough to maintain continuous operation.

Certain possible means of making the operation of the jet tube more independent of the flight speed and of reducing the flow losses were proposed and discussed.

Translation by Edward S. Shafer,
National Advisory Committee
for Aeronautics.

REFERENCE

1. Schultz-Grunow, F.: Forsch. Ing.-Wes., Bd. 13, Heft 3, 1942, p. 125.

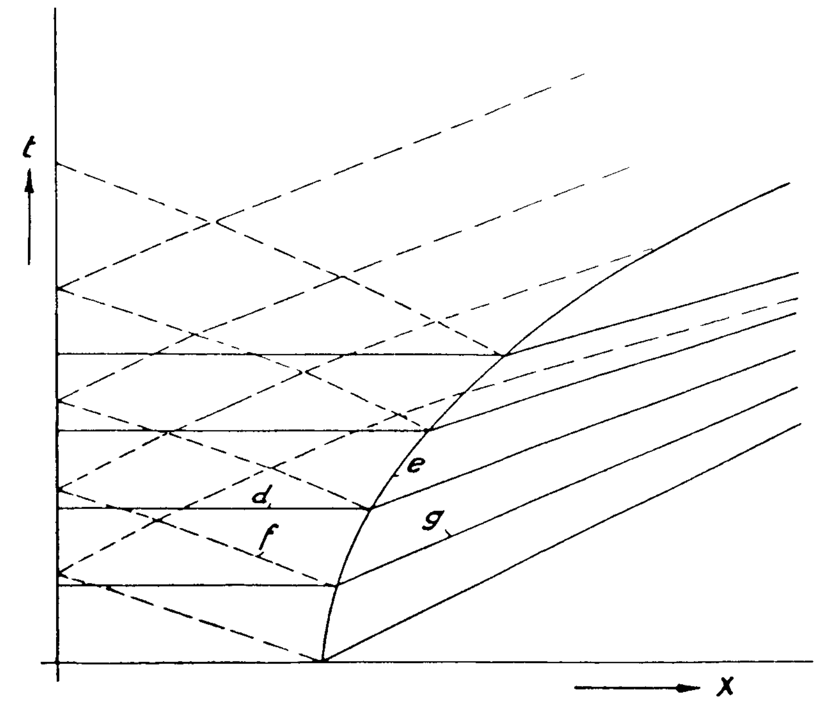


Figure 1'. - Expansion movement of column of fresh charge.
d Stages of pressure increase
e Line of propagation of fresh-charge boundary
f, g Waves of equal strength leaving fresh-charge boundary
—— Condensation waves
– – – Rarefaction waves

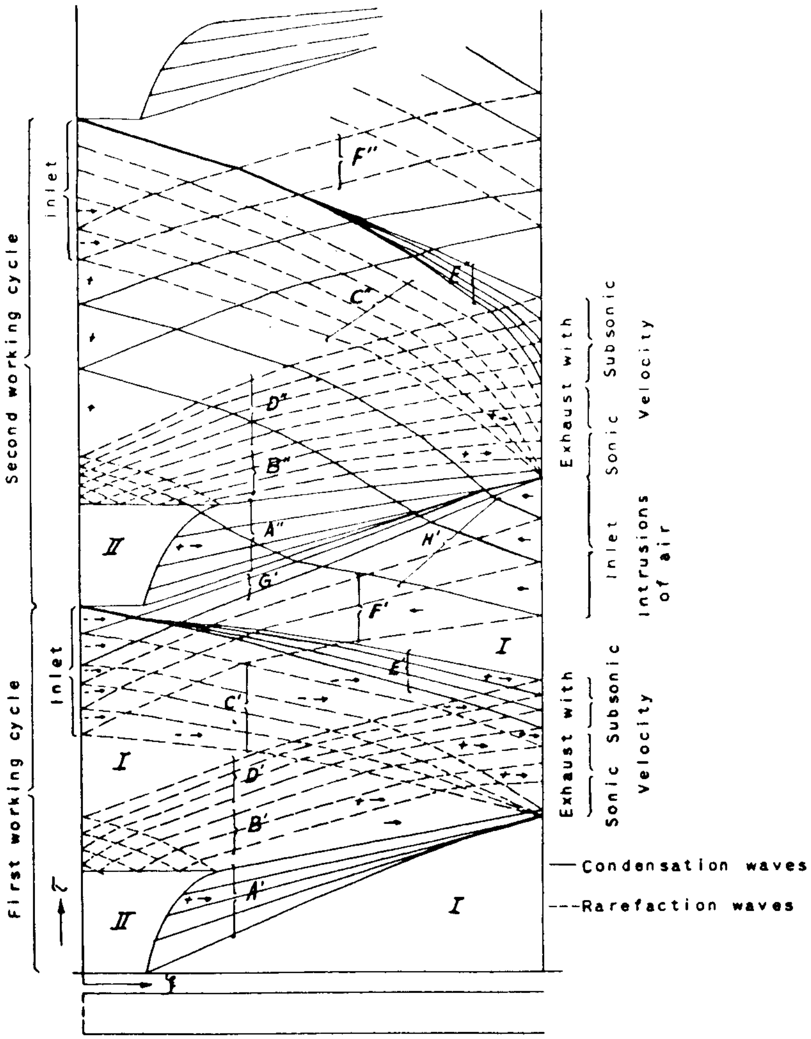


Figure 2'. - Initial phases of operation of jet tube. Wave propagation in time-distance diagram. I, atmospheric condition of rest; II, combustion.

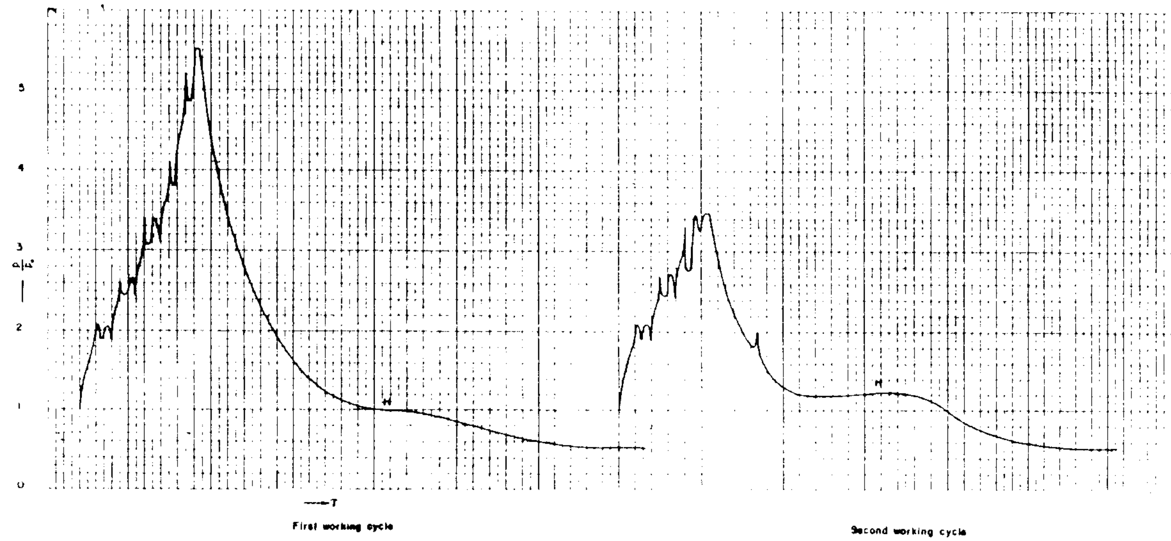


Figure 3'. - Sequence of pressures at inlet end. (A $10\frac{1}{8}$-by 21-in. print of this figure is attached.)

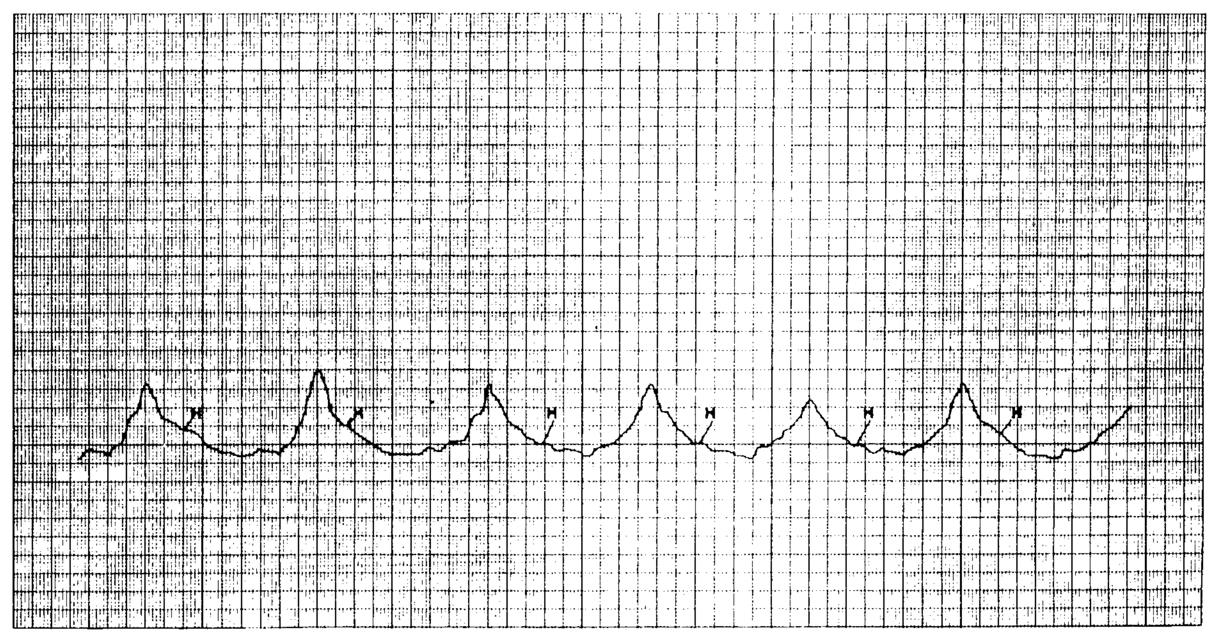


Figure 4'. - Experimentally observed sequence of pressures at inlet end obtained by Paul Schmidt.

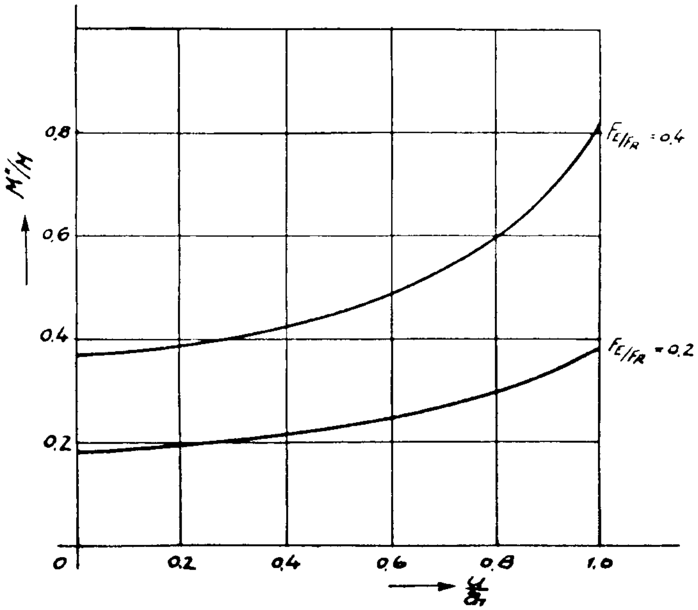


Figure 5'. - Ratio of newly indrawn quantity of fresh charge M'' to quantity originally present M' as function of flight speed u.

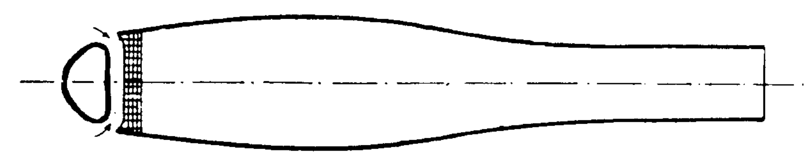


Figure 6'. - Jet tube having cap in front to reduce effect of impact pressure on valves.

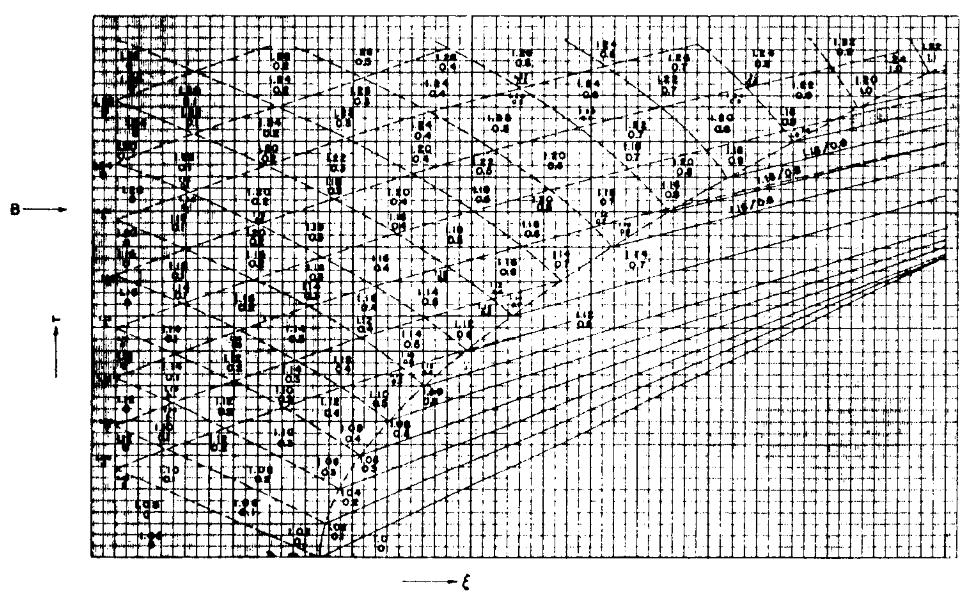


Figure 7'. - Diagram of wave propagation. Combustion. (An $11\frac{1}{2}$-by $15\frac{1}{4}$-in. print of this figure is attached.)

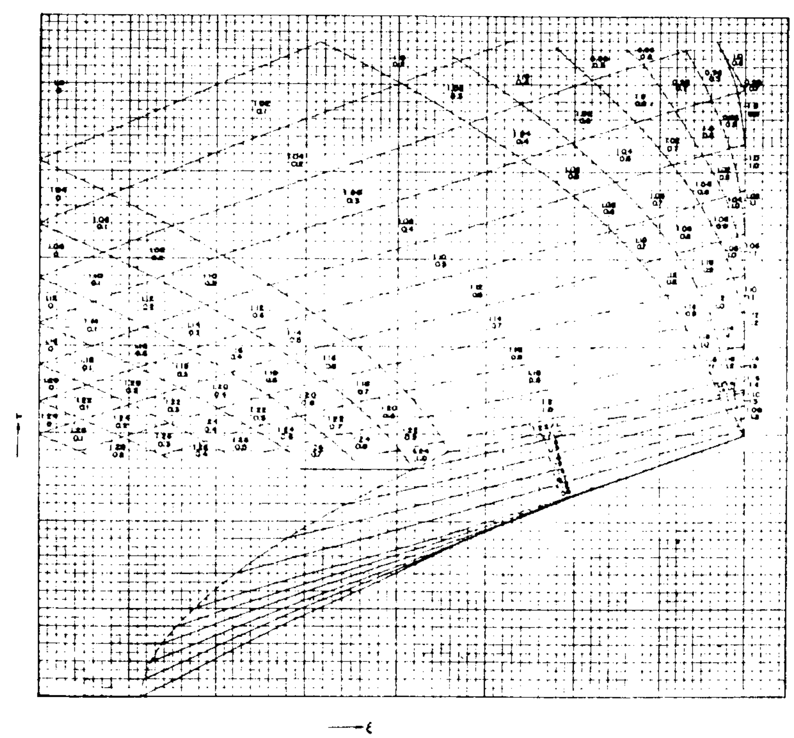


Figure 8'. - Diagram of wave propagation. First working cycle, exhaust. (A $17\frac{1}{2}$-by 18-in. print of this figure is attached.)

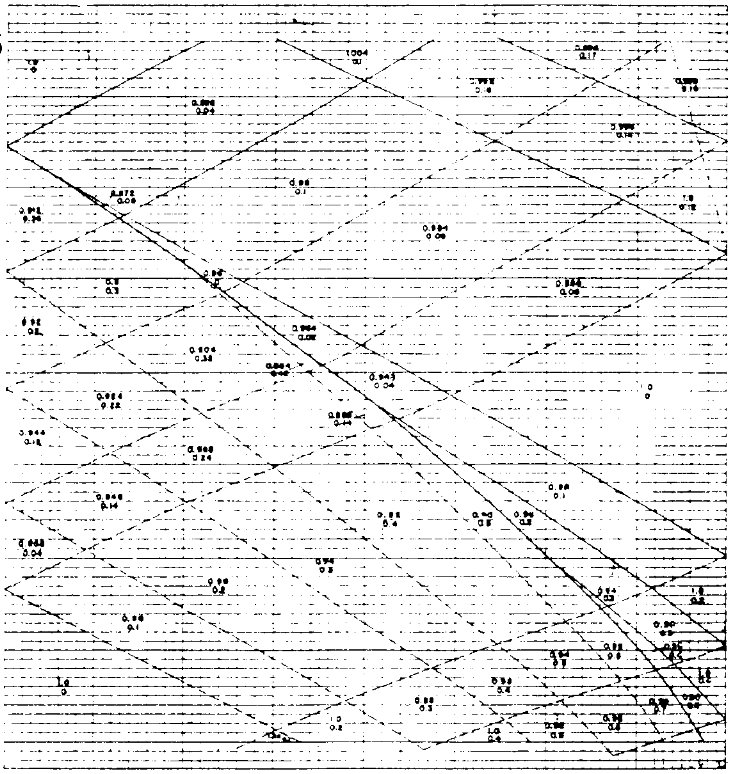


Figure 9'. – Diagram of wave propagation. First working cycle, inlet. (A $16\frac{1}{2}$-by 18-in. print of this figure is attached.)

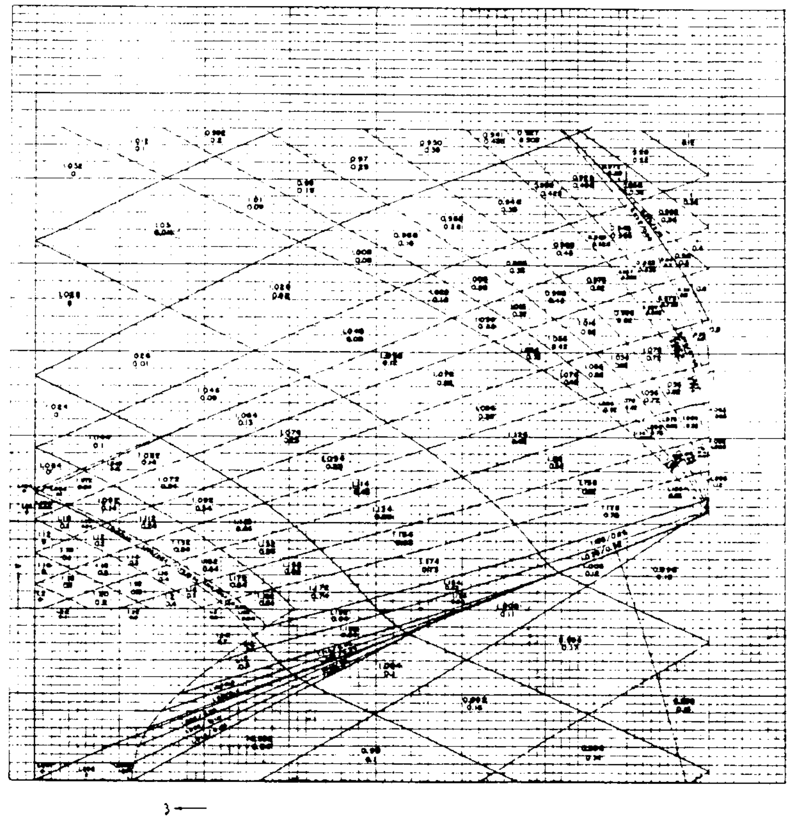


Figure 10'. – Diagram of wave propagation. Second working cycle, exhaust. (A $17\frac{1}{2}$-by 18-in. print of this figure is attached.)

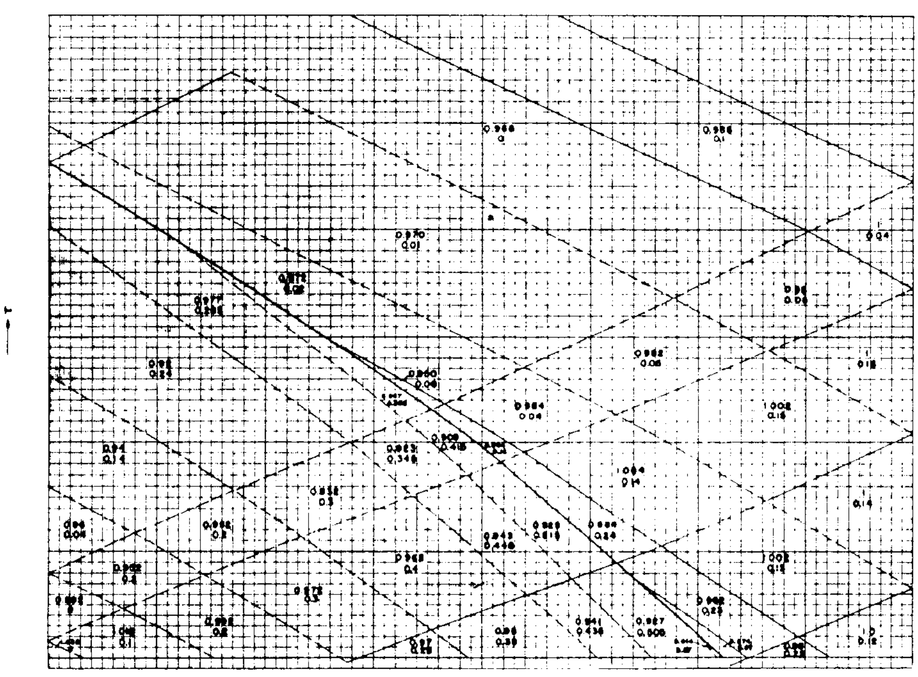


Figure 11'. – Diagram of wave propagation. Second working cycle, inlet. (A $13\frac{1}{2}$-by $16\frac{1}{2}$-in. print of this figure is attached.)

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

MEMORANDUM REPORT E243
1948?

for the

Bureau of Aeronautics, Navy Department

A PRELIMINARY EVALUATION OF THE EXPLOSION JET-PROPULSION ENGINE

By J. C. Sanders

SUMMARY

The theoretical sea-level performance of an explosion jet-propulsion engine similar to the one used in the German flying bomb was computed to show the effects on performance of heat added and supercharging and a comparison was drawn between the performance of the explosion jet-propulsion engine and the constant-pressure jet-propulsion engine. **The explosion jet-propulsion engine was found to be more efficient than the constant-pressure jet-propulsion engine at compressor pressure ratios below 3.0 when the maximum gas temperature of the constant-pressure engine is 1600° F. With more efficient compressors and higher gas temperatures, however, the constant-pressure jet-propulsion engine is more efficient. The compressor for the constant-pressure jet-propulsion engine absorbs more power than the compressor for the explosion jet-propulsion engine when the two engines develop the same power.**

INTRODUCTION

The most widely known and successful jet-propulsion engine consists of a centrifugal compressor, a constant-pressure combustion chamber, and a turbine to extract enough energy from the hot gas to drive the compressor. (See reference 1 and fig. 1(a).) The cycle efficiency of this type of engine is limited indirectly by a comparatively low maximum permissible gas temperature of 1600° F at the turbine to prevent destruction of the turbine blades. Furthermore, a highly efficient compressor capable of producing a pressure ratio of 4 or more is required.

A method of circumventing these limitations of the turbine and compressor is to use an explosion jet-propulsion engine in which the air is inducted at low pressure into a vessel, exploded to generate a high pressure, and expelled through a nozzle. (See fig. 1(b).) This type of propulsion engine was patented by Marconnet in 1909 and later patented by Schmidt in 1931 (reference 2). The Schmidt patent describes the type of engine used by the Germans to propel their

ASSUMED CONDITIONS FOR CALCULATIONS

The calculations were made for the supercharged engine shown in figure 1(c), assuming a compressor efficiency of 70 percent and a compressor driving motor having a thermodynamic cycle efficiency of 60 percent. The assumed efficiencies of the turbine and the compressor in the constant-pressure jet-propulsion engine were 80 and 70 percent, respectively. No other losses were considered.

The following operating conditions were assumed:

Altitude . . . . Sea level
Atmospheric temperature, °F 100
Velocity of aircraft, miles per hour 400
Gas temperature before turbine, °F 1600

The combined efficiency of the explosion jet-propulsion engine was computed for a range of heat supplied per pound of air (fuel-air ratio) from 0 to 750 Btu per pound. These calculations were made for compressor pressure ratios of 1, 2, 3, 4, and 8. For comparative purposes, the efficiency of the constant-pressure jet-propulsion engine was computed for the same compressor pressure ratios.

Calculations were also made to compare the thrust powers obtained from a given size compressor as first used with constant-volume combustion and second, with constant-pressure combustion. A maximum heat input of 750 Btu per pound was assumed for the constant-volume combustion and a maximum gas temperature of 1600° F was assumed for the constant-pressure combustion. A range of compressor pressure ratios from 1 to 8 was investigated.

The symbols used in the computations are given in appendix A. Details of the computations for the explosion jet-propulsion engine are described in appendix B and for the constant-pressure jet-propulsion engine, in appendix C.

DEFINITIONS OF TERMS

Cycle efficiency η_{cycle} is the ratio of the net work of the thermodynamic cycles to the heat absorbed from the fuel by the working fluid, including the fluid used by the auxiliary engine driving the compressor. The efficiency of combustion is thus excluded from the calculations.

flying bombs against England. Mention is also made of this type of engine in reference 3.

The explosion type of engine is not limited in cycle temperatures and may use a fuel-air ratio to give the maximum temperature possible in the combustion of air; furthermore, it does not require a compressor, although a compressor improves its performance.

This report presents the results of calculations showing the performance of an explosion jet-propulsion engine. The effects of supercharging on cycle efficiency and propulsive efficiency were computed and a comparison of the sea-level performance of the supercharged explosion jet-propulsion engine and the constant-pressure jet-propulsion engine was made. A schematic diagram of the supercharged engine is shown in figure 1(c).

The computations were made at the NACA Aircraft Engine Research Laboratory, Cleveland, Ohio, during February and March 1944, as a part of an investigation of means of increasing the cycle efficiencies of jet-propulsion engines.

DESCRIPTION OF THE EXPLOSION JET-PROPULSION ENGINE

The simple type of explosion jet-propulsion engine used in the German flying bombs is shown diagrammatically in figure 1(b). A combination of the dynamic pressure resulting from the forward motion of the flying bomb and the inertia effects of the previous charge being expelled from the combustion chamber serves to induce a charge of fresh air through the intake valve at the front. Fuel is sprayed into the combustion chamber and a spark plug ignites the mixture. The increasing pressure closes the valve and rises to a high value, which causes a rapid expulsion of the charge rearward and thereby generates the propulsive thrust. An idealized sketch of a captured German flying bomb using this type of propulsion engine is shown in figure 2.

Inspection of the thermodynamic cycle of the explosion jet-propulsion engine shows that both the cycle efficiency and the power can be increased by supercharging. Consequently, consideration was given to the more elaborate explosion jet-propulsion engine shown in figure 1(c). In this type of engine, a centrifugal supercharger delivers compressed air to a battery of combustion chambers provided with intake valves and possibly with exhaust valves. An auxiliary engine drives the supercharger.

Propulsive efficiency η_p is the ratio of the useful propulsive work to the net work added to the fluid by the jet-propulsion engine.

Combined efficiency η is the ratio of the useful propulsive work to the heat absorbed from the fuel by the working fluid, including the fluid used by the auxiliary engine driving the compressor. The combined efficiency may be obtained by multiplying the cycle and propulsive efficiencies, provided that they can be evaluated.

Blowdown is the portion of the explosion cycle following combustion during which enough gas is expelled from the combustion chamber to permit the pressure in the explosion chamber to fall to the exhaust back pressure.

ESTIMATED PERFORMANCE OF THE EXPLOSION JET-PROPULSION ENGINE

The combined efficiencies of the explosion jet-propulsion engine at sea level and a speed of 400 miles per hour are shown in figure 3 for compressor pressure ratios of 1, 2, 3, 4, and 8. The maximum combined efficiency is about 10 percent. In general, the combined efficiency is less at maximum heat input than at half the maximum heat input, although this difference is small. At low supercharger pressure ratios, the combined efficiency falls off rapidly at very low values of heat input.

A comparison of the efficiencies of jet-propulsion engines using constant-volume combustion and constant-pressure combustion is shown in figure 4. The constant volume of the explosion jet-propulsion engine is more efficient at compressor pressure ratios below 3.0. A single-stage centrifugal supercharger is capable of producing a pressure ratio of 4. Inspection of figure 4 shows that the explosion jet-propulsion engine with a compression ratio of 4 is about 4 percent less efficient than a constant-pressure jet-propulsion engine having the same compressor pressure ratio.

The ratio of powers obtainable from these two types of engine with the same size compressors is shown in figure 5. The power obtainable with constant-pressure combustion is less than the power obtainable with constant-volume combustion and, of course, falls to the very low efficiency of an engine using the dynamic pressure of the air when the compressor pressure ratio is 1.

The advantage of the higher efficiency obtained by supercharging is obtained at the cost of a large and heavy engine to drive the supercharger. Figure 6 shows that the power required by the compressor engine exceeds the net thrust power when the supercharger pressure ratio is 2.0 or greater. It therefore appears that supercharging is practical only for low pressure ratios.

4 not 2 see Fig. 6 and #3 p. 6

DISCUSSION

Reliability of calculations. - The trends of the performance curves shown in figures 3, 4, and 5 were expected because examination of the working cycles of the explosion and constant-pressure jet-propulsion engines shows that the efficiency of the constant-pressure cycle must be zero with a compressor pressure ratio of 1, assuming no dynamic compression. Furthermore, the explosion cycle permits expansion of most of the gas from a much higher pressure than is available in a constant-pressure cycle and therefore usually has a higher cycle efficiency. In spite of the lower cycle efficiency of the constant-pressure engine, its combined efficiency is expected to exceed the efficiency of the explosion jet-propulsion engine under conditions of optimum design because the propulsive efficiency of the explosion jet-propulsion engine is very low as a result of the high jet velocities at the beginning of blowdown.

The assumption was made in these calculations that the blowdown to atmospheric pressure was complete. Consequently, the velocity of discharge during scavenging was negligible. It is possible, however, that the explosion vessel would blow down to supercharger pressure and the residual gases would be expelled at a velocity corresponding to the pressure drop from supercharger pressure to atmospheric pressure. Calculations of combined efficiencies with this type of operation (shown in fig. 3 for comparison with performance with complete blowdown) show lower efficiencies at high heat inputs but higher efficiencies at very low heat inputs than in the case of complete blowdown.

Other factors that may lower the performance of the explosion jet-propulsion engine are energy losses in charging and scavenging the combustion chamber, poor scavenging of the combustion chamber, poor combustion, and degeneration of the ideal cycle resulting from expulsion of gas through the nozzle before combustion is complete. Insufficient data are available to evaluate the significance of these factors and they were therefore not considered in the calculations.

Charging. - The charging process in the explosion jet-propulsion engine may be similar to the uniflow-type two-stroke cycle engine in which inlet and exhaust valves are at opposite ends of the cylinders.

ρ_b weight of charge remaining in vessel at end of any assumed blowdown process, (slug)

ρ_1 density of air at atmospheric conditions, (slugs)/(cu ft)

T_o temperature of gas in vessel after completion of combustion, °F

T_1 temperature of air entering supercharger, °F absolute

T_2 temperature of air leaving supercharger, or at end of compression, °F absolute

T_3 temperature of air entering compressor, accounting for adiabatic temperature rise to stagnation, °F absolute

T_4 temperature of gas approaching turbine, °F absolute

V_o velocity of aircraft with respect to earth, (ft)/(sec)

V_f velocity of escaping gas relative to airplane, (ft)/(sec)

w_b thrust work developed during blowdown, (ft-lb)/(cu ft)

w_h work equivalent of heat added to fluid in auxiliary engine driving the compressor, or added to gas during combustion, (ft-lb)/(lb of fluid passed through nozzle of jet), or (ft-lb)/(slug)

w_o total work available in adiabatic expansion. (ft-lb)/(slug)

w_N net thrust work, (ft-lb)/(slug of air)

w_t work abstracted by turbine to drive compressor, (ft-lb)/(slug)

Z effective thrust coefficient

APPENDIX B

CALCULATIONS FOR THE EXPLOSION JET-PROPULSION ENGINE

The equation for the combined efficiency of a jet-propulsion engine is:

$$\eta = \frac{\text{Net thrust work}}{\text{Energy supplied by fuel}} \qquad (1)$$

The net thrust work is the thrust work developed by the vessel blowing down minus the work required to bring the charge air to a zero velocity with respect to the airplane. The energy supplied to the working fluid includes the heat added to the air in the combustion chamber and the heat added to the working fluid in the auxiliary motor driving the compressor. In computing the thrust work developed during blowdown, it is necessary to determine the maximum amount of heat to be supplied to the cycle by the fuel. These computations therefore describe in turn (1) a derivation of the expression for thrust work of a vessel blowing down, (2) the work of the fluid in the engine driving the supercharger, (3) a determination of the maximum heat supplied, and (4) the final equation for the efficiency of an explosion jet-propulsion engine.

Thrust developed by a vessel blowing down. - The vessel is considered to be filled with its charge and combustion complete, with the gas pressure P_o and temperature T_o. The vessel then blows down to atmospheric pressure p_4. If the volume of the vessel is 1 cubic foot the weight of the initial charge is ρ_o slugs. The thrust force at any instant is

$$F = V_f\left(\frac{d\rho}{dt}\right)$$

The thrust work developed during the expulsion of the charge $d\rho$ is

$$dw_b = FV_o dt$$

$$= V_o I_V$$

Substituting the value of F and integrating the left side gives

$$I_V = \int_{\rho_o}^{\rho_b} V_f d\rho \qquad (2)$$

If it is assumed that the gas obeys the general gas law, $PV = wRT$, then

$$V_f = 223.7\left(\frac{c_p\,P_o}{R\,\rho_o}\right)^{\frac{1}{2}}\sqrt{\frac{\gamma}{\gamma-1}\left[1-\left(\frac{p_1}{P_o}\right)^{\frac{\gamma-1}{\gamma}}\left(\frac{\rho_o}{\rho}\right)^{\gamma-1}\right]} \qquad (3)$$

Substituting the value of V_f from equation (3) in equation (2) and simplifying and converting to impulse per slug of gas in cylinder before blowdown yields

$$I_{slug} = 223.7\left(\frac{c_p\,P_o}{R\,\rho_o}\right)^{\frac{1}{2}} Z \qquad (4)$$

where Z is the effective thrust coefficient defined as

$$Z = \int_{\rho_o}^{\rho_b}\sqrt{\left(\frac{\rho}{\rho_o}\right)^{\gamma-1} - \left(\frac{p_1}{P_o}\right)^{\frac{\gamma-1}{\gamma}}}\; d\,\frac{\rho}{\rho_o} \qquad (5)$$

Values of Z obtained by arithmetic integration of equation (5) are shown in figure 7.

The value of P_o was determined as follows:

$$P_o = p_4 \times abc$$

The value of a was determined by

$$a = \frac{h}{c_v T_2} + 1$$

The value of c was determined by

$$c = \frac{P_4 + \left(\frac{1}{2}\rho_4 V_o^2 \beta\right)}{2117}$$

The value of ρ_o is calculated from

$$\rho_o = \rho_4 (bc)^{0.714}$$
$$= 0.00250\, b^{0.714}$$

When the proper values are substituted, equation (4) becomes:

$$I_{slug} = 53.0\left(\frac{P_o}{b^{0.714}}\right)^{\frac{1}{2}} Z \qquad (9)$$

where

$$I_o = 2520b + 28.2b^{0.714}h \qquad (10)$$

The net thrust work is

$$w_N = V_o I_{slug} - V_o^2$$
$$w_N = \left[53.0 \times 587\left(\frac{P_o}{b^{0.714}}\right)^{\frac{1}{2}} Z\right] - 587^2$$

Work of fluid in the engine driving the supercharger. - The work of the fluid in the engine driving the supercharger was computed by dividing the energy required for adiabatic compression by the adiabatic efficiency of the compressor and the cycle efficiency of the compressor driving engine:

$$w_h = \frac{\gamma}{\gamma-1} RT_1 \left(b^{0.286} - 1\right) \frac{1}{(\eta_{ad})(\eta_{cycle})} \qquad (11)$$

The value of T_1 is equal to

$$T_1 = T_3 (c)^{0.286} = 1.19\, T_3$$

and

$$w_h = 262,000 \left(b^{0.286} - 1\right) \qquad (12)$$

Maximum heat supplied in explosion. - It was assumed that the combustion in an explosion jet-propulsion engine would be similar to combustion in an internal-combustion engine using spark ignition and that the maximum heat added to a pound of mixture in the spark-ignition engine would likewise be the maximum added in a jet-propulsion engine. Indicator-card analysis in reference 6 showed that the maximum heat was delivered to the charge with a fuel-air ratio of 0.082. Analysis of the indicator card for this fuel-air ratio showed that 750 Btu were added per pound.

Final equation for efficiency of jet-propulsion engine. From equation (1)

$$\eta = \frac{I_{slug} V_o - V_o^2}{32.2(778h + \text{supercharger work})}$$

Substituting the proper values from equations (9) and (12) gives:

$$\eta = \frac{966\left(P_o/b^{0.714}\right)^{\frac{1}{2}} Z - 10,700}{778h + 262,000\left(b^{0.286} - 1\right)} \qquad (13)$$

APPENDIX C

CALCULATIONS FOR CONSTANT-PRESSURE JET-PROPULSION ENGINE

The efficiency of the constant-pressure jet-propulsion engine is given by the equation

$$\eta = \frac{(w_o - w_t)\eta_p}{w_h} \qquad (14)$$

Total work available in adiabatic expansion. - The total work w_o available in adiabatic expansion is given by the usual equation for the work of an air motor:

$$w_o = \frac{\gamma_1}{\gamma_1 - 1} RT_4 \left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{\gamma_1 - 1}{\gamma_1}}\right] \qquad (15)$$

Work abstracted by turbine. - The work abstracted by the turbine is given by the equation:

$$w_t = \frac{\gamma_2}{\gamma_2 - 1} RT_3 \left\{\left[\left(\frac{p_3}{p_2}\right)^{\frac{\gamma_2 - 1}{\gamma_2}} - 1\right] \frac{1}{\eta_c \eta_t}\right\} \qquad (16)$$

Propulsive efficiency. - The propulsive efficiency was computed by the use of equation (14) of reference 7, which is:

$$\eta_p = \frac{2}{1 + \sqrt{1 + \epsilon - K}}$$

Heat added during combustion. - The heat added during combustion was computed by the following equation:

$$h = 778\, c_p\, (T_o - T_2) \qquad (17)$$

The temperature T_2 is computed by adding to the ambient atmospheric temperature (assumed to be 560° F absolute) the adiabatic temperature rise to stagnation before the intake to the compressor and the temperature rise through the compressor.

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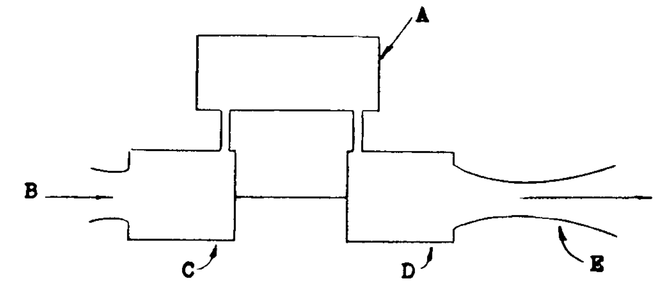
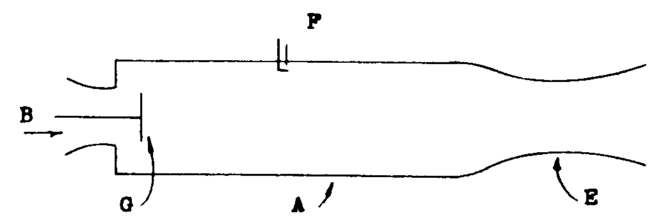
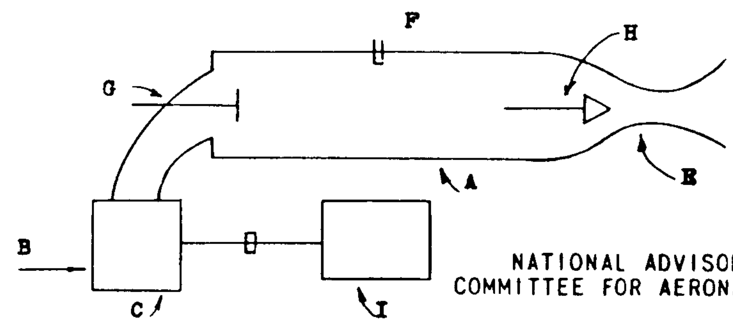


(a) A jet-propulsion engine using constant-pressure combustion.



(b) A simple explosion jet-propulsion engine.



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(c) A supercharged explosion jet-propulsion engine.

A Combustion chamber
B Air intake
C Compressor
D Turbine
E Nozzle

F Spark plug
G Intake valve
H Exhaust valve
I Motor for driving supercharger

Figure 1. - Schematic diagrams of jet-propulsion engines using constant-pressure and constant-volume combustion.

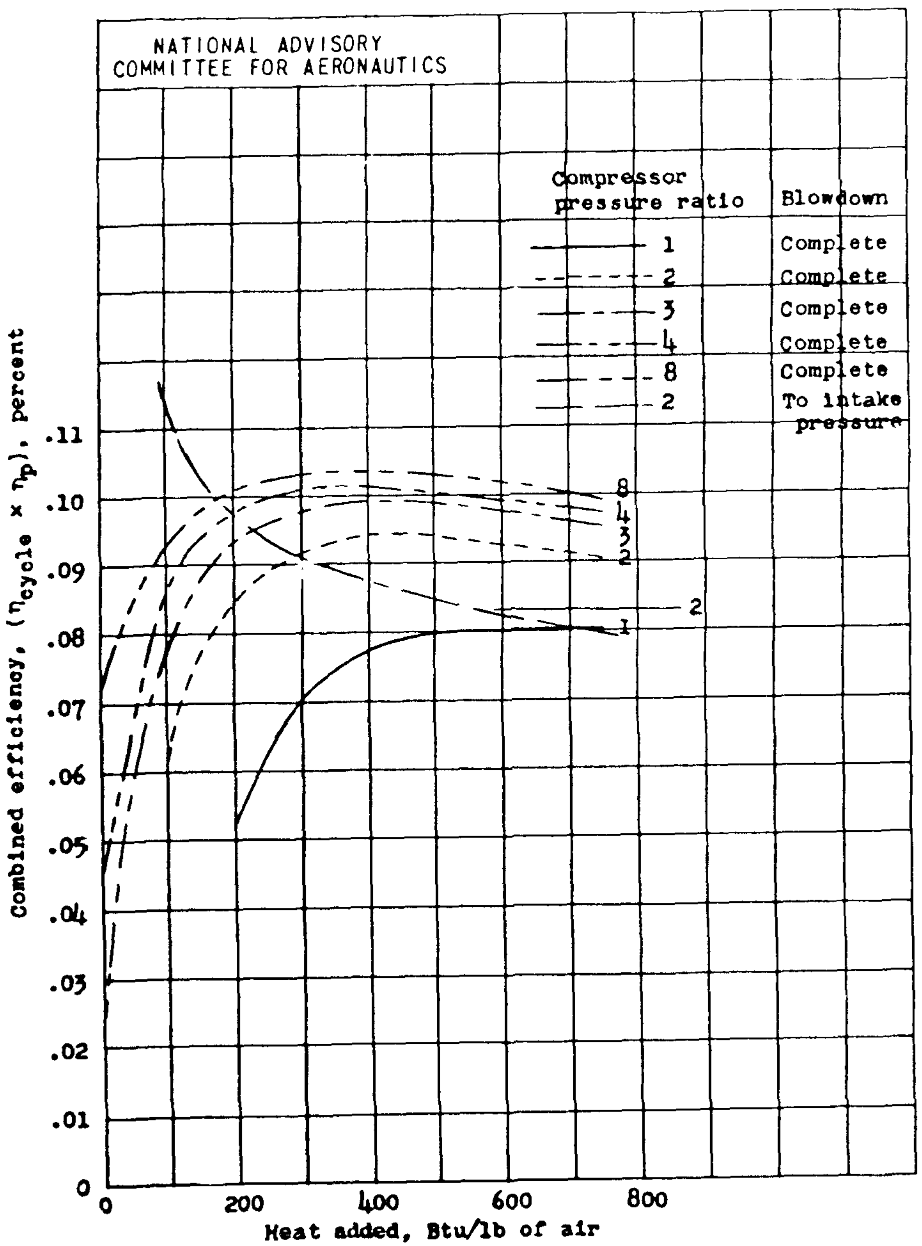


Figure 3.- Effect of heat input and pressure ratio of the compressor on the combined efficiency of an explosion jet-propulsion engine.

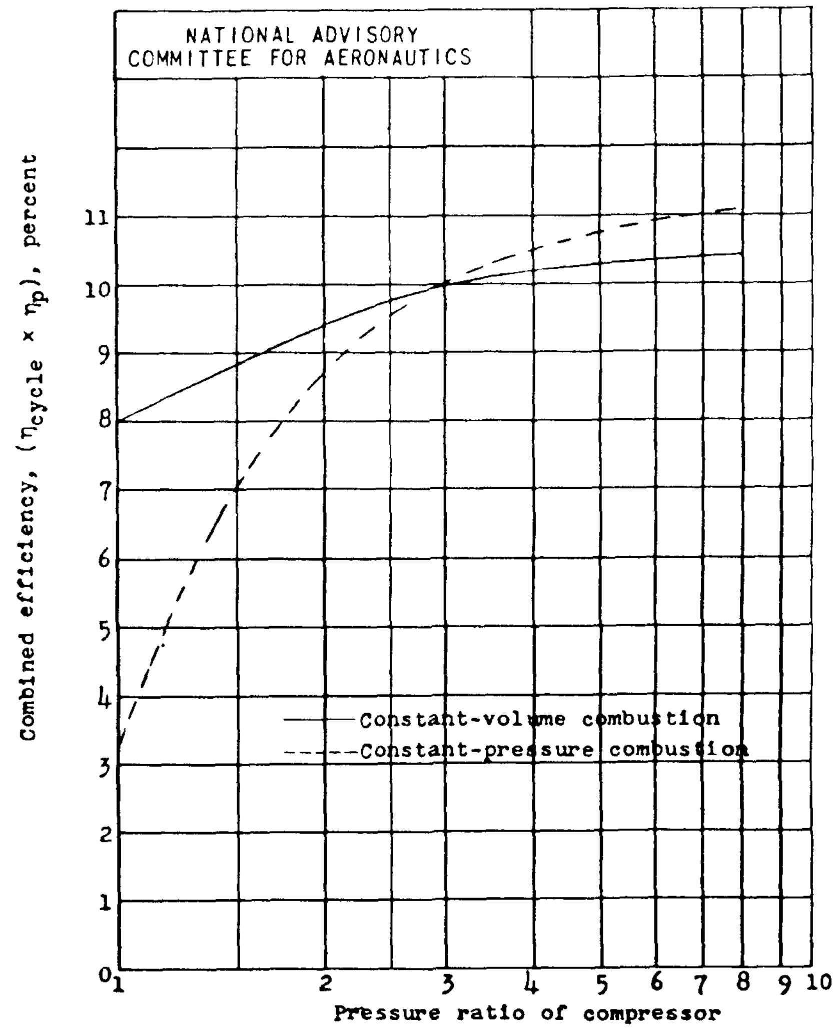


Figure 4.- Comparison of efficiencies of jet-propulsion systems using constant-pressure combustion and constant-volume combustion. Each system at sea level; constant-volume-combustion engine using heat input for maximum economy.

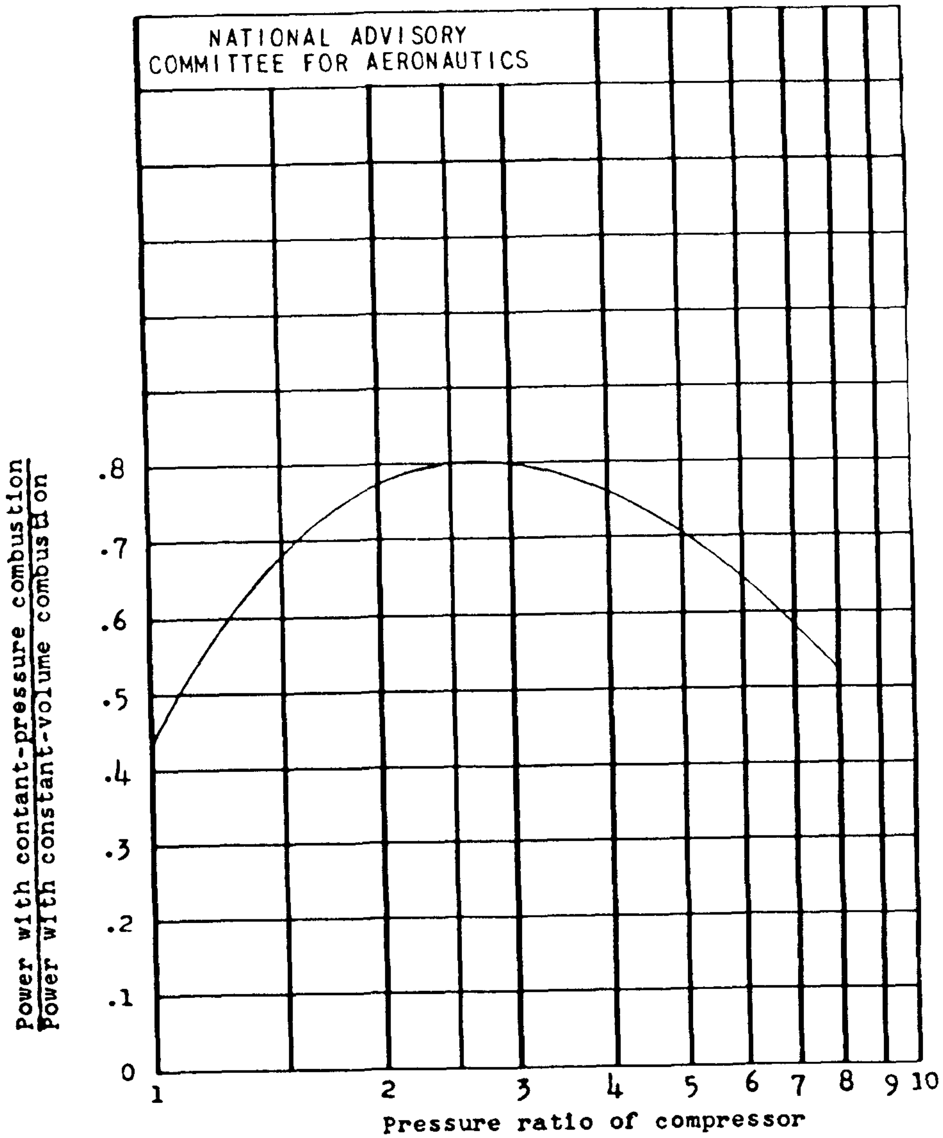


Figure 5.- Comparison of the thrust power obtainable from a given compressor in jet-propulsion cycles using constant-pressure and constant-volume combustion.

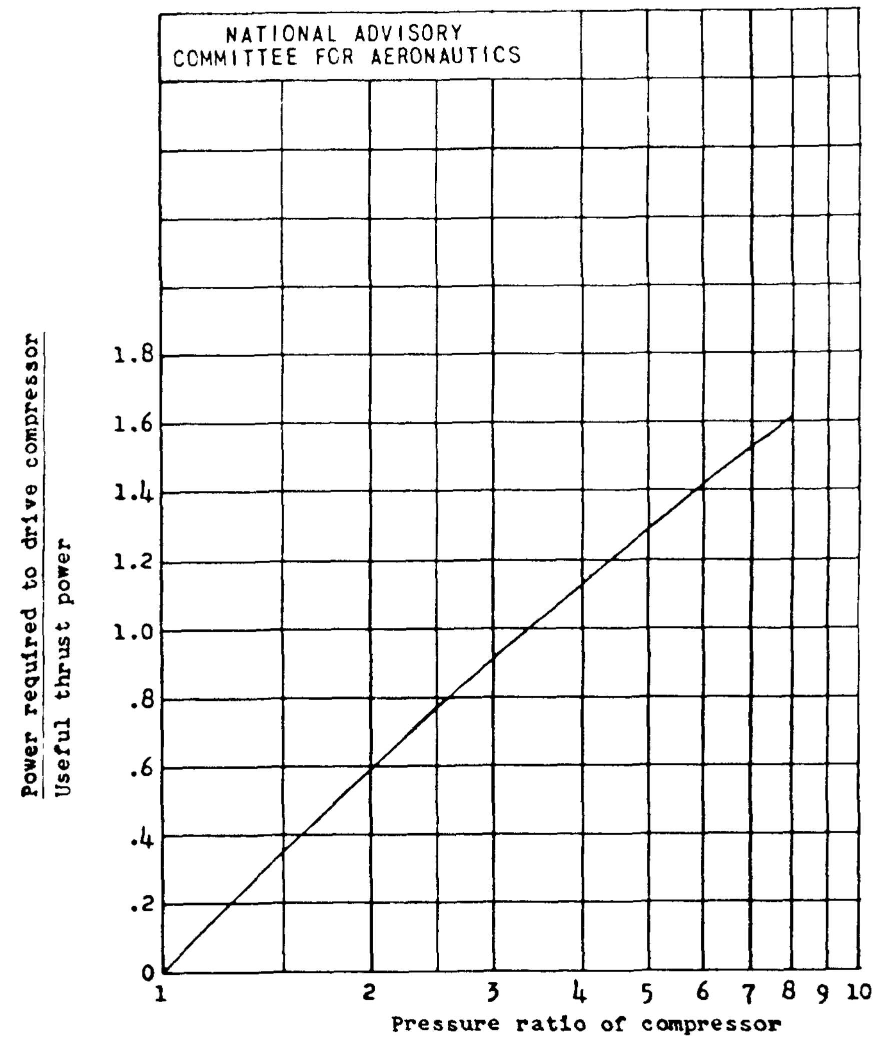


Figure 6.- Relation of useful thrust power of an explosion jet-propulsion engine to the power required to drive its compressor.

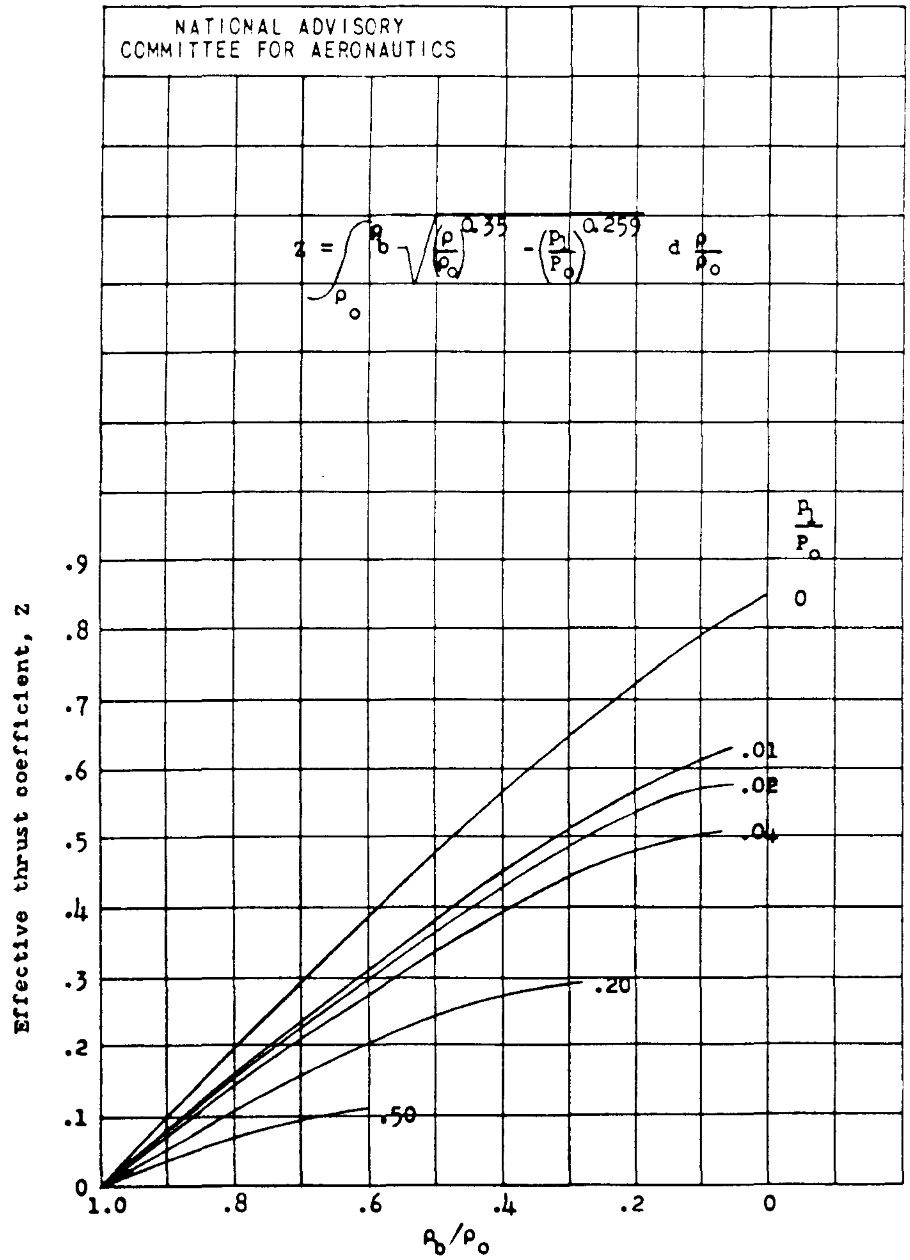


Figure 7.- Values of the effective thrust coefficient z for a vessel periodically discharging from high pressure.

Pulsating Jet Engines—A Survey of the Development of Ignition

By P. SCHMIDT. (From *Zeitschrift des VDI*, Vol. 92, No. 16, June 1, 1950, pp. 393-399, 14 illustrations.)

A CHARACTERISTIC feature of the pulsating jet engine is the automatic ignition which is propagated through the combustible mixture as a flame front of very high velocity. Work on the development of this type of ignition—about which little has been published—was started in 1931 in an attempt to solve the problem of pulsating high frequency combustion at constant volume in tubes open at one end. For this purpose, rates of flame propagation far in excess of those occurring in internal combustion engines were required.

Single ignition experiments were based on Wendlandt's finding that detonation of normally non-detonating gas mixtures could be induced by hard shock waves. A small volume of a volatile liquid fuel, such as ethyl ether, was added to the explosion chamber *a* (see Fig. 1), compressed air of 5 atmospheres was blown in, and the mixture was ignited by sparking plug *e*. The explosive blast fractured cellophane diaphragm *c*, initiating a shock wave in combustion tube *b* filled with the combustible mixture. Under suitable conditions, detonating combustion was obtained with a variety of fuels.

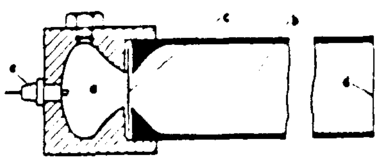


Fig. 1. Equipment for shock wave experiments.
(a) Explosion chamber ; (b) combustion tube ; (c) cellophane diaphragm ; (d) paper diaphragm ; (e) sparking plug.

The effect was similar when the initial shock—the bursting of the cellophane diaphragm—was caused by the introduction of a compressed non-burning gas mixture into chamber *a*. Although, on expansion, the temperature of the compressed gas tended to drop very low, ignition was obtained as long as the pressure had been above a critical value. Careful consideration led to the conclusion that detonation could have been caused only by the shock wave. The efficiency of energy transformation in these experiments with tubes open at one end was determined as 70 per cent.

In the next stage of development, single igniting shock waves were produced mechanically by a free piston arrangement. This was followed by the periodically operating device shown diagrammatically in Fig. 2. Ports *b* permit the combustible mixture to enter cylinder *a*. Free piston *e* moving to the right forces out the portion of the mixture not required for ignition through flap valves *d* into the combustion tube (not shown) to the right of the small opening *f*. On further movement of the piston to the right, the mixture trapped in the cylinder is compressed adiabatically to self-ignition temperature. The sudden rise in pressure forced a shock wave through opening *f*, causing the ignition in

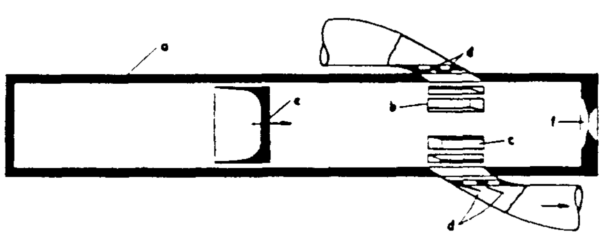


Fig. 2. Pulsating ignition apparatus.
(a) Cylinder ; (b) intake ports ; (c) exhaust ports ; (d) flap valves ; (e) free piston ; (f) opening.

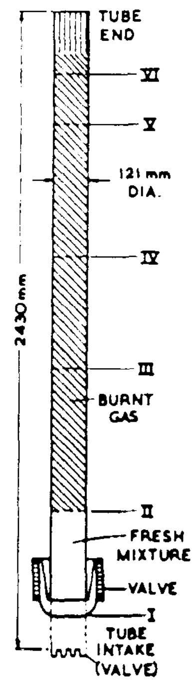


Fig. 3. Cylindrical experimental tube with pressure pick-ups located at I to VI.

the tube to the right of the opening, and also returned the piston to its initial position.

It was found in 1937 that in pulsating tube operation under suitable conditions, ignition was automatic, so that the apparatus for producing shock waves was no longer required. However, it was important for automatic pulsating ignition that the area of the combustion air intake and the sectional area of the exhaust end of the tube should be carefully related. With a tube shown diagrammatically in Fig. 3, pressure oscillograms were taken at several points. They are reproduced in Fig. 4. It can be seen that the explosive pressure wave (line *a*) is reflected back at the tube end as suction wave *b*. This depression is followed by a short, but very steep, pressure rise, which at point *c*, hits the front of new combustible mixture, starting combustion immediately. A new small shock wave (*c*) is initiated.

The pulsating jet tubes used in these early experiments (described here for the first time) had only a

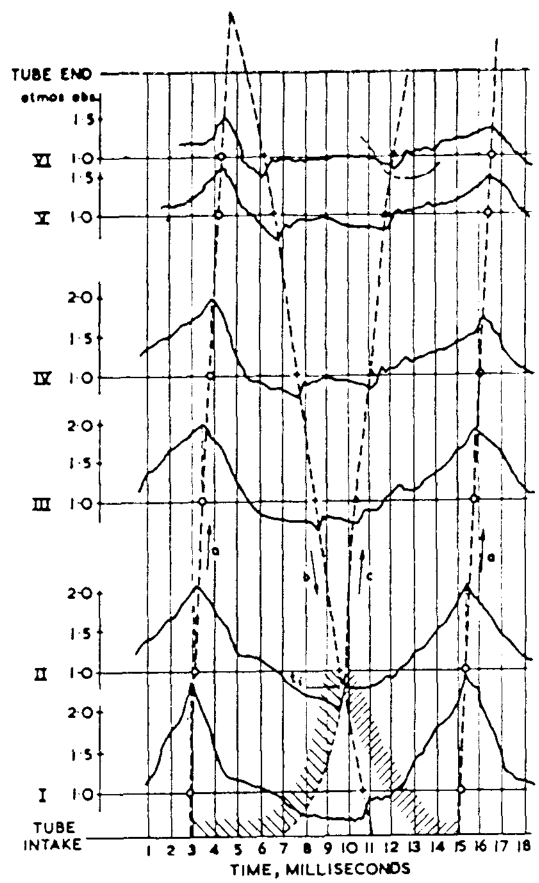


Fig. 4. Variation of gas pressure with time in the tube shown in Fig. 3.

moderate output, operating at maximum pressures between 2·5 and 3·5 atm. Later on, tubes of about 500 mm diameter operated continuously with maximum pressures of 5-6 atm. abs. The performance of these tubes was almost entirely independent of the fuel used. The tube diameter could be varied between 10 and 600 mm, and the shape of the tube, cylindrical or barrel type, straight or coiled, or the method of mixture formation, had no influence. Ignition was extremely reliable and was not stopped by considerable deformation of the tube or severe damage to valve or tube wall. Operation was not impaired by moderate static pressures at the exhaust end and can take place at considerable altitude.

Pulse-Jet Possibilities

A MOST interesting letter has been received from a German engineer, Herr Jos. Reder, whom many regular readers will remember as the author of a provocative article in THE AEROPLANE for December 22, 1950, on the potential applications of pulse-jets as a means of aircraft propulsion. In his most recent letter, which is printed full length hereafter, Herr Reder makes a critical review of the Escopette pulse-jet and continues with a discussion on pulse-jets.

He has studied with great interest our description of the Escopette pulse-jet in "Power Plants on Show" (Paris Aero Show, THE AEROPLANE, June 22, 1951, pages 771-772).

Obviously, he writes, there must be something wrong with the given fuel-consumption figures of 2 to 5.5 grams/kg. thrust/sec. The specific fuel/thrust per hour would thus be 7.2 to 20 kg./kgp./hr. (7.2 to 20 lb./lb. thrust/hr.) which is far too high and absolutely out of the question in comparison with any other conventional turbo-jet power plant of about 1-1.2 kg./kgp./hr. (Pimené and Palas turbo-jets). On the other hand, the second figure of 0.33 lb./lb./hr. is far too low for such pulse-jets.

Most probably the gross fuel consumption of the single pipe is 2 to 5.5 grams/sec., and with this figure the specific fuel consumption would be 2.4 to 2 kg./kgp./hr. (2.4 lb. to 2 lb./lb. thrust/hr.) with thrust variation from 3 to 10 kgp. (6.6 to 22 lb. thrust). This is quite possible if compared with the famous V.1 pulse-jet AS14 which had a very bad economy and used about 3 to 4 kg./kgp./hr.

These observations by Herr Reder are of great interest for the reason that another and independent French source gives the specific fuel consumption as 1.8 kg./kgp./hr. (1.8 lb./lb. thrust/hr.) for the Escopette compared with the figure given us at the Aero Show.

A table, worked out by Herr Reder, of jet propeller developments, shows that the Paris Aero Show Escopette is not the "dernier-cri" in pulse-jet development. In the years 1943-45 quite a lot of different pulse-jet couplings were tested and the one shown in Fig. 1 (resembling very much a double "Escopette") had about 35 to 55 per cent. better specific fuel/thrust consumption, 1.2 to 2 kg./kgp./hr. (1.2 to 2 lb./lb. thrust/hr.). The coupling of the inlet-valve-pipes alone increased the thrust about 20 to 30 per cent. and the outlet coupling, together with proper cowling, gave a further improvement of 15 to 25 per cent.

The figure shows that each single pulse-jet loads each other one (in inlet and outlet) if resonance speed frequency is hit. That means, that exactly the right ratio of pipe lenghts ($L_1/L_2 = 1/8$ to $1/12$) and the right diameters of the different pipes must be kept and the cowling volume must be in proper proportion to the volume of a single pulse-jet-pipe (about 2 or 3 times as much).

The double Escopette with a specific fuel/thrust consumption of about 1.2 kg./kgp./hr. (1.2 lb./lb. thrust/hr.) comes very near to the economy of the current small turbo-jets like the Pimené or Palas, which is pretty good considering the fact that the pulse-jet costs only a fraction of the complicated turbo-jets. Of course, Herr Reder writes, there is still that ugly noise which shakes one's teeth out of their sockets.

But he contends that this horrible noise can be minimized, if ultra-high gas swingings (i.e., ultra-high frequency surging) in very short pipes are utilized and the whole pulse-jet unit is put into the hub of a hollow single-bladed airscrew, blowing out through tip orifices.

Such pulse-jets (RE-JET) would only work if the fuel is prepared by high pressure and heat for instant ignition (detonation) in spherical combustion chambers. (Panzerfaust Effect.) In any fast-running airscrew very high pressure is available through centrifugal forces and fraction heat of flowing gases can be recuperated for fuel-preparing. (Presumably this means that the heat of the exhaust gases can be used to preheat the fuel.) Thus, well over twice as much heat energy

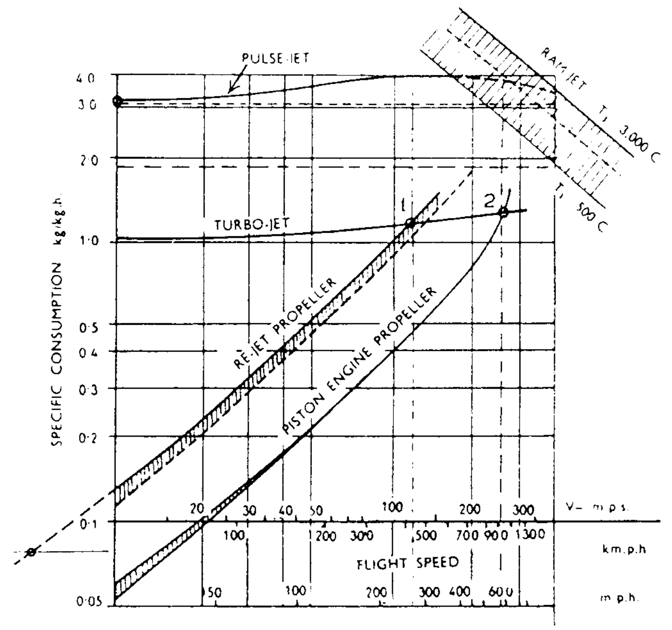


Fig. 2

can be utilized out of normal low octane fuel and plain water in pulse-jet propellers as would ever be given off by the improved double Escopette with its long pipes.

Model tests with small Jetex-propellers (with solid fuel charges of high specific fuel/thrust ratios) have proved that it is quite possible to improve the specific fuel/thrust consumption to about 7 times that of the direct rocket drive (from 29 to 4 kg./kgp./hr., 29 to 4 lb./lb. thrust/hr.) at about half the specific propeller-weight/thrust ratio (from 1-0.7 to 0.4 kg./kgp. (lb./lb. thrust)).

The figures with a Dyna-jet propeller would be much better. The specific consumption is about 0.3 kg./kgp./hr. (0.3 lb./lb. thrust/hr.) at static thrust, with single-bladed airscrews of about 0.8 metre (31.5 in.) diameter. The weight/thrust ratio is only 0.3 to 0.35 kg./kgp. (lb./lb. thrust) at the very low thrust/propeller area ratio of about 30 kgp./$m.^2$ (6 lb. thrust/sq. ft.).

The top of all these "Jetteries" would be the RE-JET-TUNNEL-PROPELLER, or ducted-fan pulse-jet, with a take-off fuel/thrust consumption as low as 0.08 kg./kgp./hr. (0.08 lb./lb. thrust/hr.), which is far better than any turbo-jet will ever have and is only about twice as much as that of a piston engine of equal power (Fig. 3).

The ratio of propeller-weight to take-off thrust of about 0.12 kg./kgp. (0.12 lb./lb. thrust) would also surpass any conventional power-plant at the same take-off thrust. Even at cruising speed with about 0.52 kg./kgp. (0.52 lb./lb. thrust) the RE-JET propeller would compare favourably with a Palas turbo-jet, because it would cost only a fraction (about 1/10 to 1/15) of the much too complicated turbos; also the pulse-jet is much easier in maintenance.

Not only does the tunnel serve as a condenser, to regain the otherwise lost water of the hot-water tip-rocket for boosted take-off thrust, it does not add to the power-plant weight because in a "Ducky-Duck" (tail-first plane, THE AEROPLANE, December 22, 1950) the tunnel is an important part of the

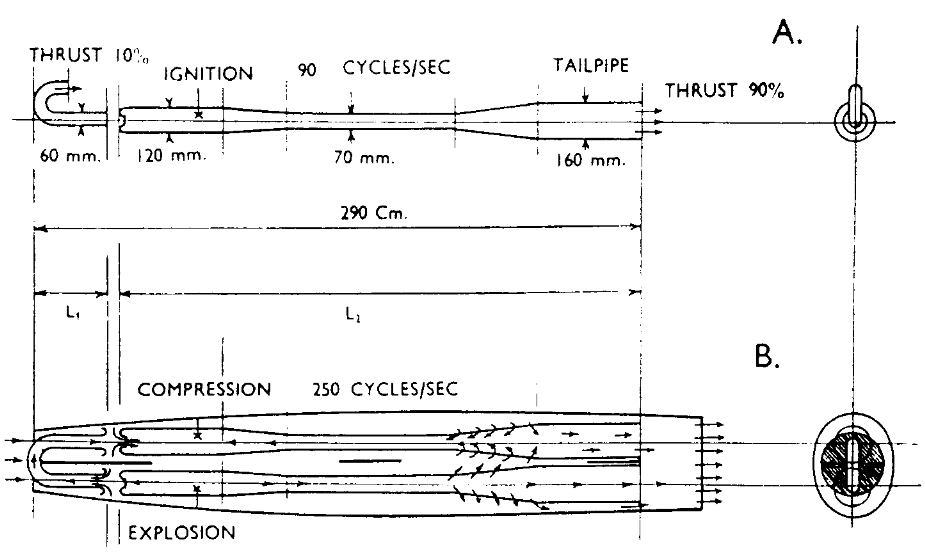


Fig. 1

ESCOPETTE PULSEJET

A Compact, Powerful Unit: Its Background History and Working Principle

This photograph shows the method of installation of six Escopette units on the Emouchet sailplanes which are being used for the initial experiments. The dry weight of each pulsejet, with equipment, is 4.8 kg. Maximum static thrust is 10 kg.

VISITORS to the last year's Paris Show flying display will recall the impressive, if ear-shattering, performance put up by the little Emouchet glider with its novel array of six Escopette Pulsejets. This unit has since been subjected to an extensive programme of trials, in the course of which over 200 flights have been made on two sailplanes, one fitted with four and the other six pulsejets. Hitherto, the Escopette has been studied primarily in the light of its capabilities in the low speed-range, where fuel consumption (at 1.8 kg/kg/hr) is particularly good. Now comes news that SNECMA are developing a number of higher-powered models for use at substantially greater speeds. As we shall, therefore, undoubtedly be hearing more of this unconventional power unit it seems timely to offer a brief history of its origin and explain the principle on which it works.

In 1670 Huygens discovered the basic functional principle of modern pulsejets, namely, that sudden evacuation of air from a cylinder will produce an atmospheric depression inside it; in piston engines this is known as the Kadenacy effect. Huygens described a propulsive machine working on this principle with the aid of gunpowder, but it was not until the dawn of the twentieth century that the first working application of the "pulsating" combustion-chamber appeared. These engines made use of Huygens's depression effect to permit successive refuellings of a chamber in which combustion at approximately constant volume was taking place. In this latter respect pulsejets are related directly to piston engines and differ from turbojets and rocket motors, in which combustion continues at constant pressure.

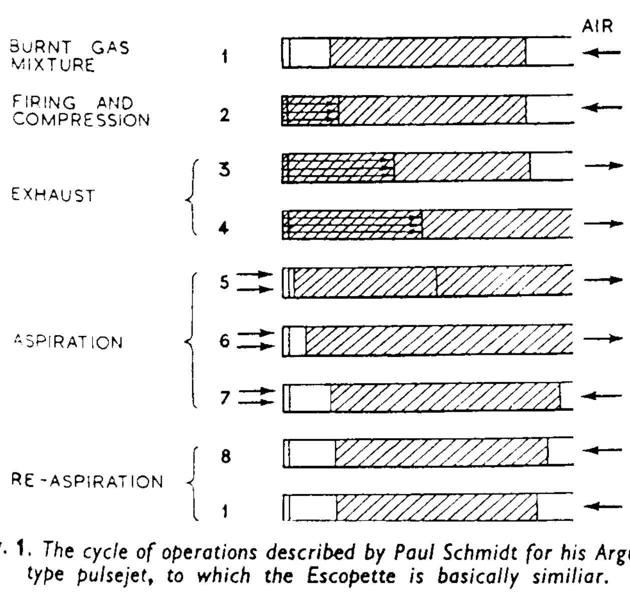


Fig. 1. The cycle of operations described by Paul Schmidt for his Argus-type pulsejet, to which the Escopette is basically similiar.

Holzwarth, in 1908, began to study a chamber, with automatic valves, feeding a gas turbine. In 1909 Marconnet was working on models which were very similar to the modern Escopette. As early as 1910, Caravodine built a gas turbine comprising a combustion chamber without pre-compression and giving an output which would be considered acceptable even today. Esnault-Pelterie also played an important part in these experiments.

Arising from these early studies, many investigations were made in various countries around 1930 but only those of Paul Schmidt resulted in a successful production model, later to become the Argus unit in the V.1 flying bomb. A passage from one of Schmidt's memoirs will serve to describe its principle of operation, which, incidentally, is almost identical with that of the modern Escopette (Fig. 1 is a schematic diagram showing the various phases of operation) :—

"The main tube is provided with an inlet valve for combustible air. This valve [represented by a double line in the diagram] allows the fuel/air mixture to enter and prevents the return of the exhaust gas after the mixture is ignited, that is to say, during the generation of pressure inside the tube. At the top of the diagram the tube is shown at the end of a cycle; near the inlet valve is a charge of fresh mixture. In the remainder of the tube there are burnt gases from the previous combustion, with a body of air near the outlet orifice. The second illustration shows the stage immediately after ignition. Small arrows indicate the effect of pressure due to the combustion; this effect takes place very rapidly so that the gas column in the tube is pushed towards the rear, as shown in the third and fourth illustrations. Due to inertia, this movement continues even after motive pressure has ceased. The depression which is then set up in the tube, particularly near the valve, draws a new column of mixture into the tube (as indicated by the arrows in stages 5, 6 and 7).

"Air also enters the tube through the exhaust outlet (Schmidt's 're-aspiration') and is forced out again in the next cycle. Thus the tube functions at pre-determined resonance frequency. Not only is there an automatic intake of fresh air, but also, automatic ignition takes place at high combustion rates." It should be noted that

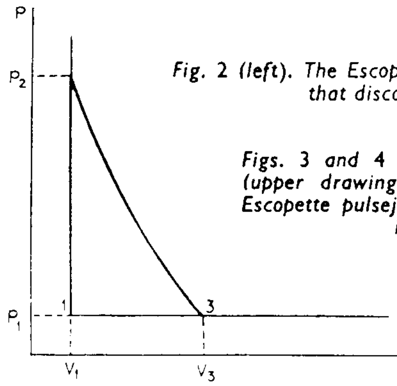
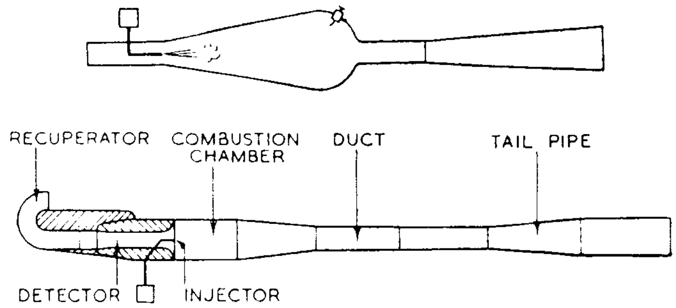


Fig. 2 (left). The Escopette's thermodynamic cycle is that discovered by Lenoir.

Figs. 3 and 4 (right). The Marconnet unit (upper drawing) of 1909 and the modern Escopette pulsejet; the principle of operation is similar.



ESCOPETTE PULSEJET . . .

there is no actual explosion, but rather a high-speed combustion of the order of 30 m/sec.

Thermodynamic Principle.—The thermodynamic cycle is that used by Lenoir in his gas engine which functioned without pre-compression. Ignition occurred during the period of induction. As illustrated in Fig. 2, this cycle comprises an important increase in pressure at constant volume, followed by an adiabatic expansion down to the inlet pressure. It has been calculated that the maximum pressure attained can reach seven times the initial pressure, P_0; the theoretic efficiency under these conditions being 28 per cent. In point of fact, the pressure attained is actually less because of the width of the openings of the combustion chamber and because the fresh charge is incomplete, the combustion being relatively slow. It is of the order of two atmospheres, or even less at smaller throttle-openings.

Propulsive Efficiency.—The thermodynamic efficiency of a pulse-jet is extremely low, but it is still able to retain its place as a power unit because its propulsive output is relatively high. This is due to the fact that each discharge is also followed by the induction of a certain volume of air into the exhaust end of the tail-pipe. This air is then expelled at the next successive combustion.

Thus the kinetic energy produced is applied to a larger mass of air, which, although having the effect of lessening the exit-velocity nevertheless increases the momentum (and therefore the thrust) of the discharge gases.

Details of the Escopette.—The Escopette is a pulsejet of the Marconnet type (Figs. 3 and 4). It has no inlet valves and gas movement is regulated only by varying the dimensions of its duct. It comprises the following parts : a detector, combustion chamber, main body or duct, tail-pipe, recuperator, fuel injector and igniter plug. The detector is the name given to the intake unit, which has the general form of a venturi with a bias towards the passage of fluid in one direction only. Although outwardly it appears to be a simple component, development of the detector has, nevertheless, necessitated numerous tests and constitutes the most original feature of SNECMA pulsejets. Apart from eliminating the need for mechanical valves it also has the following advantages. Having no precise frequency of its own it can be adapted to harmonize with the main apparatus; it has a long life; fuel injection is easily made into the detector inlet opening; and a good combustible mixture is obtained even from continued injection at low pressures.

Counteracting these advantages, however, is the fact that the detector is not airtight and at each cycle produces a considerable back-pressure. As this back-pressure exerts itself in a forward direction, a curled "walking-stick" tube called a recuperator and shaped something like the cup of a Pelton turbine, is provided to reverse this movement. Its effect is to increase the output of the power unit. There is actually a gap between the recuperator and the detector to permit the entry of "induction air."

The functions of the body and tail-pipe are similar to those of normal pulsejets. The igniter plug located in the combustion chamber is only used for starting, as in normal running the re-igniting of successive charges is automatic.

The Escopette operates on petrol of any octane value, injected at pressures varying from 0.3 to 1.4 atm. At fuel-pressures outside these limits the unit will not function. Thrust varies from 3 to 10 kg at consumption figures which can be from 2 to 5 g/sec, according to the type of injector used. It has been discovered that, if two or three pulsejets are mounted close together, the output of each unit is decreased by ten per cent for the same specific fuel consumption.

For starting, the spark is provided by a battery- or magneto-operated igniter plug. An injection of compressed air must be made at a pressure of 3 to 5 atm through a jet (4 to 10 mm in diameter) mounted in the entrance of the detector tube and directed towards the combustion chamber. These "squirts" of air must be frequent, but of short duration. Fuel injections are made at the same time as the air so as to avoid the possibility of a residue of liquid fuel accumulating in the combustion chamber. Should ignition not take place, the fuel must be cut off immediately and pure air injected to blow out surplus fluid. Electric contact is switched off as soon as the pulsejet begins to function.

A number of safety precautions have to be observed when these units are fitted to an aircraft. For example, they must be located as far as possible away from any inflammable material, while wing-surfaces near to the body of a pulsejet should be covered with a sheet of polished aluminium to reflect the very considerable heat generated by the combustion chambers.

A typical installation for four Type 3340 Escopettes (as fitted in an Emouchet sailplane) is shown in Fig. 5. All the requisite fuel- and air-pressure cocks (for starting), together with the two main throttles (controlling fuel pressure and air pressure feed to fuel tank), are located in the cockpit, but outside assistance is required when ground-starting to direct the jets from a compressed-air bottle into the intake of the pulsejets. For starting, the air pressure to the fuel tank must be regulated to 1.5 atm and fuel pressure to 0.75 atm by the two throttle controls referred to above. Maximum duration at ground level is 10 min at full throttle, i.e. a fuel pressure of 1 atm.

In flight, starting or restarting is easily accomplished at a speed of approximately 75 km/hr; all the pilot needs to do, having set his throttle at the starting position, is to switch on the ignition and open the fuel cocks.

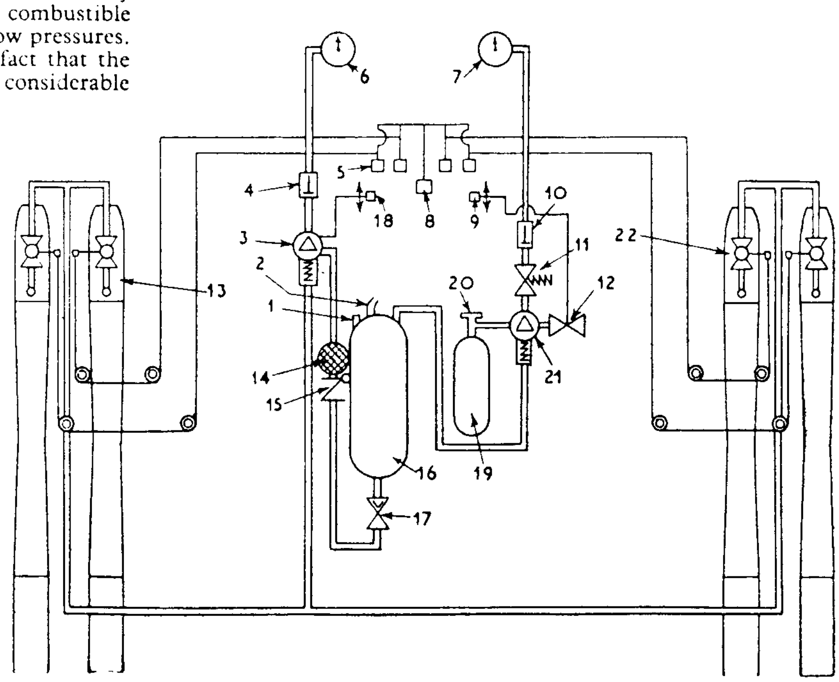


Fig. 5. Fuel circuit of a typical four-unit installation, showing the positions of controls and valves.

1. Fuel-tank filler cap.
2. Excess-pressure vent.
3. Throttle valve.
4. Relay to fuel pressure gauge.
5. Individual taps for fuel regulator valves.
6. Fuel pressure gauge.
7. Air pressure gauge.
8. Master fuel-feed cock.
9. Air control throttle.
10. Relay to air-pressure gauge.
11. Safety valve (air, 4 atm).
12. Emergency pressure release.
13. Fuel regulator valve.
14. Fuel filter.
15. Non-return valve.
16. Fuel tank (20 litres).
17. Main fuel cut-off.
18. Fuel-control throttle.
19. Compressed air bottle (1 litre at 80 atm).
20. Main air cock.
21. Air regulator valve.
22. Fuel injector.

Das Schmidtrohr

Von Dipl.-Ing. **H. Lembcke**, München

Nach einer kurzen Aufführung der in den westlichen Ländern für das Schmidtrohr verwendeten verschiedenartigen Bezeichnungen werden seine Wirkungsweise, die erreichten und zukünftig möglichen Leistungsbereiche sowie einige Anwendungen aufgezeigt.

Bezeichnung

In deutschen und ausländischen Veröffentlichungen sind in letzter Zeit wiederholt Strahlgeräte beschrieben worden, die mit periodischen Verbrennungen in sehr schneller Folge arbeiten. Es handelt sich dabei entweder um Antriebseinrichtungen, z. B. für Flugzeuge, oder auch um Geräte, bei denen die Wärme oder die Energie des Gasstrahls in anderer Weise ausgenutzt wird.

Entsprechend den verschiedenen Anwendungszwecken findet man im Schrifttum verschiedenartige Bezeichnungen für solche Geräte. In Deutschland wurden u. a. die folgenden Benennungen verwendet: Verpuffungsstrahlrohr, Verpuffungsrohr, Schubrohr, Pulsotriebwerk, Aero-Resonator, Stoßbrenner, Schwingfeuergerät, Schwingrohr. In ausländischen Veröffentlichungen sind Bezeichnungen wie Pulsating Jet Engine, Pulse-Jet, Dyna-Jet, Pulso-réacteur, Escopette usw. eingeführt worden.

Bei allen diesen Geräten handelt es sich um die Anwendung des von *Paul Schmidt*, München, geschaffenen Verfahrens der pulsierenden Verbrennung in einem einfachen Rohr mit bis dahin in Verbrennungsmaschinen unerreicht hohen Zündgeschwindigkeiten. Er hatte seinen nach diesem Verfahren arbeitenden Antrieb, den er im Jahre 1930 erfunden und in den Jahren bis 1940 schon zu großen Einheiten entwickelt hatte, bis zum Jahre 1944 als Strahlrohr bezeichnet [1]. Als dann für den daraus hervorgegangenen Strahlantrieb der V1 in den beteiligten Kreisen teilweise unzutreffende Namen verwendet wurden, ergab sich für das Reichsluftfahrtministerium die Notwendigkeit, eine einheitliche Bezeichnung festzulegen. Nach eingehender Untersuchung der Zusammenhänge wurde angeordnet, daß der Strahlantrieb der V1 als Argus-Schmidtrohr zu bezeichnen sei. Das Wort Schmidtrohr — in dieser Schreibweise — kennzeichnet dabei das neuartige Betriebsverfahren, während das mit Bindestrich davorgesetzte Wort Argus zur Angabe der Fertigungsfirma dient.

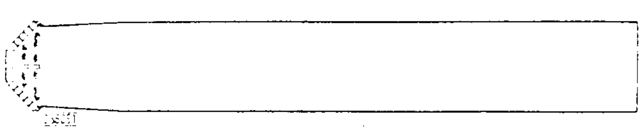


Bild 1. Gesamtansicht.

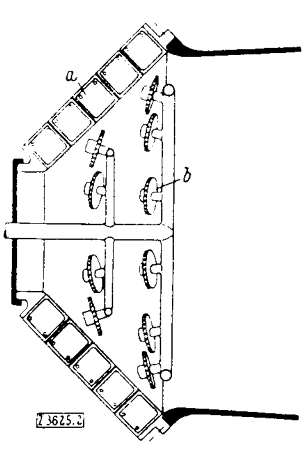


Bild 1 und 2. Schmidtrohr SR 500.
a Ventilklappen
b Brennstoffzerstäuber

Bild 2. Eintrittsseite des Rohres.

Eigenschaften und Leistungsbereich

Grundsätzlicher Aufbau

Bild 1 und 2 zeigen die Grundform des Schmidtrohres, wie sie seit dem Jahre 1937 von *P. Schmidt* auf dem Prüfstand untersucht wurde. Das dargestellte Rohr SR 500 mit 510 mm Dmr. und rd. 3,5 m Länge war im Jahre 1939 in der Erprobung.

Es hat sich gezeigt, daß man diese Grundform bei gleichbleibenden Betriebseigenschaften weitgehend abwandeln kann. So kann man das Verhältnis von Länge zu Durchmesser beträchtlich verschieden wählen; das Rohr kann man sich verengend oder sich erweiternd ausführen oder sogar in Windungen legen. Auch die Gestaltung des Lufteinlaßventils bietet verschiedenartige Möglichkeiten, z. B. kann an Stelle der üblichen Ventilklappen auch ein dynamisches Ventil verwendet werden, wie es bereits im Jahre 1942 von *P. Schmidt* erprobt wurde. Ein solches Ventil arbeitet ohne bewegte Teile auf der Grundlage, daß sein Strömungswiderstand beim Rückströmen größer als beim Einströmen ist. Eine weitere bemerkenswerte Eigenschaft des Schmidtrohres ist seine Unempfindlichkeit gegenüber den verschiedenen Brennstoffen.

Verbrennungsvorgang

Die Leistung des Schmidtrohrs beruht auf der im Rohr stattfindenden Gleichraum-Verbrennung. Die Tatsache, daß in einem einseitig offenen Brennraum eine Gleichraum-Verbrennung erreicht wird, erklärt sich aus den ungewöhnlich hohen Zündgeschwindigkeiten, die im Schmidtrohr auftreten. In Rohren mittlerer Größe ist die Zündgeschwindigkeit rd. 500 m/s. Bei so großen Geschwindigkeiten kann es sich nicht um eine Zündung auf Grund von Wärmeleitung und Turbulenz handeln. Die Zündung wird vielmehr durch eine Stoßwelle verursacht, die mit sehr hoher Geschwindigkeit vom offenen Rohrende zurückläuft [2; 3].

Brennkammerleistung

Die hohe Zündgeschwindigkeit führt zu überraschend hohen Leistungen offener Brennräume, wie es in eindrucksvoller Weise z. B. durch britische Messungen bestätigt wird [4]. Die übliche Kennzeichnung der Brennkammerleistung durch Angabe der Umsetzung in Wärmeeinheiten je Stunde und Volumeinheit des Brennraums ist jedoch für periodisch arbeitende Brennräume nicht geeignet. Eine so gekennzeichnete Brennkammerleistung ändert sich nämlich mit der Periodenzahl; da die Periodenanzahl der Brennraumlänge umgekehrt verhältnisgleich ist, würden sich bei kleinen Brennkammerausführungen außerordentlich hohe Werte ergeben. Zweckmäßiger ist es daher, die Brennkammerleistung auf den mittleren Querschnitt der Brennkammer zu beziehen; die sich so ergebenden Werte sind von der Größe der Brennkammerausführung unabhängig. In dieser Weise aus den britischen Messungen berechnete Bezugswerte liegen bei etwa $90 \cdot 10^6$ kcal/m² h. Sie stimmen mit den Werten überein, die von *P. Schmidt* an seinen selbstansaugenden Versuchsrohren in den Jahren 1937 bis 1945 festgestellt wurden.

Schubkennwerte und Wirkungsgrad

Als Kennwerte für die Leistung eines Schmidtrohrs bei der Verwendung als Flugzeugantrieb dienen besonders das Verhältnis von Baugewicht zu Schubkraft im Stand, das Verhältnis von Brennstoffverbrauch zu Schub im Stand und der Gesamtwirkungsgrad bei verschiedenen Fluggeschwindigkeiten.

Bis zum Jahre 1942 war von *P. Schmidt* bei Rohren großer Leistung (der größte Standschub eines Rohres betrug 750 kg) ein Verhältnis von Baugewicht zu Schub im Stand von etwa 0,3 kg/kg erreicht worden; allerdings wurde dabei keine Entwicklungsarbeit nach kleinem Baugewicht hin betrieben.

In Bild 3 ist ein Oszillogramm eines Prüfstandversuches mit einem Rohr nach Bild 1 wiedergegeben (Verbrennungsdruck, Schub und Brennstoffverbrauch). Die günstigsten Verbrauchszahlen mit Rohren dieser Bauart lagen etwa bei 2,5 kg Brennstoff je kg Schub und Stunde. Für die mit „Escopette" bezeichnete Bauart wird ein Brennstoffverbrauch von 1,8 kg/kg h bei einem Baugewicht von 0,51 kg/kg angegeben [5]. Die von anderen Bauarten

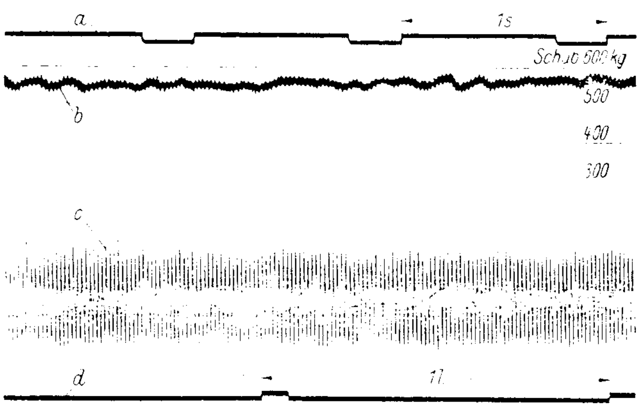


Bild 3. Oszillogramm eines Schmidtrohrs SR 500 (Versuch Nr. 40 899).

a Zeit *c* Druck im Verbrennungsrohr
b Schub *d* Brennstoffverbrauch

bekanntgewordenen Werte liegen in der gleichen Größenordnung. Bei der Verwendung des Schmidtrohres zum Antrieb von Drehflügeln konnte sogar ein Verhältnis von Rohrgewicht zu Schub von nur 0,24 kg/kg erreicht werden (vgl. Abschnitt über Anwendungen).

Der Gesamtwirkungsgrad des Rohres im Fluge betrug bei der V1 etwa 4%; dabei kam aber der Schub des Rohres wegen der aerodynamisch ungünstigen Anordnung nur zum Teil als Nutzschub zur Geltung.

Verbesserungsmöglichkeiten

Mit den bisher bekannt gewordenen Ausführungen sind die Grenzen der Leistung von Schmidtrohren noch nicht erreicht; vielmehr läßt sich auf Grund weiterer Untersuchungen übersehen, daß noch erhebliche Verbesserungen möglich sind. Der Gütegrad der Verbrennung und die Luftvorlagerung im Fluge, die beim Rohr der V1 nur sehr kleine Werte erreichten, können durch einige grundsätzliche Änderungen in der Bauart des Rohres wesentlich verbessert werden. Man kann damit rechnen, daß mit solchen Verbesserungen und als Folge einer Leichtbau-Entwicklung ein Baugewicht von 0,1 bis 0,2 kg je kg Schub im Stand und ein Brennstoffverbrauch von etwa 1 kg je kg Schub und Stunde erreicht werden können.

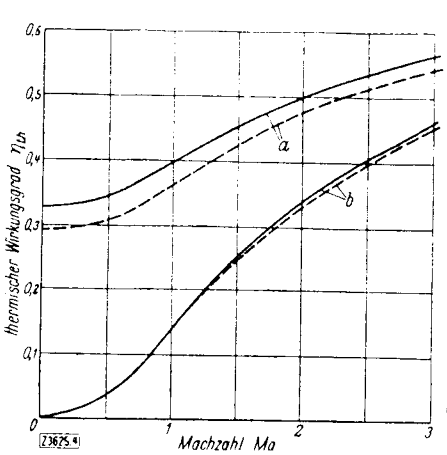


Bild 4. Thermischer Wirkungsgrad η_{th} des Schmidtrohrs (Kurven *a*) und des Staustrahlrohrs (Kurven *b*) in Abhängigkeit von der Machzahl Ma.

Ma = 1 entspricht einer Fluggeschwindigkeit von rd. 1200 km/h; ausgezogene Kurven für Flughöhe h = 11 km, gestrichelte Kurven für h = 0 km

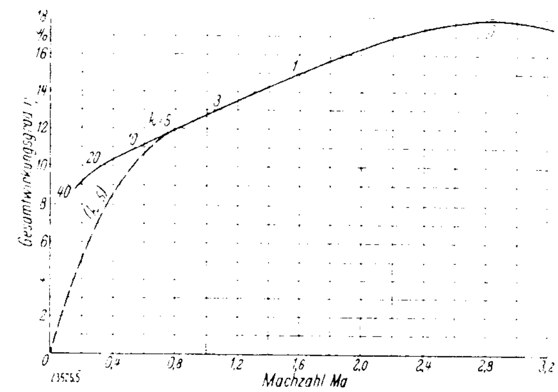


Bild 5. Gesamtwirkungsgrad η des Schmidtrohrs in Abhängigkeit von der Machzahl Ma.

Gestrichelte Kurve wahrscheinlicher Verlauf bei kleinen Ma
k Massenverhältnis von beschleunigter Luft zu verbranntem Gemisch

Für die Beurteilung der Wirkungsgrade eines Schmidtrohrs im Fluge ist die durch den Staudruck erreichbare Verdichtung des Gemisches wesentlich. Bei Betrieb im Stand ist für die periodische Gleichraum-Verbrennung im Schmidtrohr der thermische Wirkungsgrad des Lenoir-Prozesses maßgebend, der rd. 30% beträgt. Infolge der Verdichtung durch den Flugstau steigt der thermische Wirkungsgrad, und zwar in verschiedenen Höhen etwas verschieden (entsprechend der Lufttemperatur). Bild 4 zeigt die Abhängigkeit des thermischen Wirkungsgrades η_{th} von der Machzahl Ma. Dabei ist beim Berechnen des thermischen Wirkungsgrades die Temperaturabhängigkeit der spezifischen Wärme berücksichtigt worden. Zum Vergleich enthält das Bild auch den thermischen Wirkungsgrad für ein Staustrahlrohr („Ramjet") mit stetiger Gleichdruck-Verbrennung. Man sieht, daß die Gleichraum-Verbrennung des Schmidtrohrs erheblich günstigere Werte als die Gleichdruck-Verbrennung beim Staustrahlrohr ergibt.

Bild 5 gibt eine Übersicht über die zu erreichenden Gesamtwirkungsgrade des Schmidtrohrs in Abhängigkeit von der Machzahl. Die mit k bezeichneten Werte geben das Verhältnis der kolbenartig beschleunigten zusätzlichen Luftmasse zur Masse des verbrannten Gemisches an. Es ist als nicht ganz sicher anzusehen, ob die für kleine Machzahlen eingetragenen recht hohen Werte k ≈ 5 der Luftvorlagerung erreicht werden können. Deshalb ist die Kurve für k = 5 gestrichelt bis Ma = 0 fortgesetzt.

Bild 6. V1-Haube.

Bild 7. Speicherhaube.

Bild 6 und 7. Sichtbarmachung der Haubeneinströmung mittels Fadenbüschels.

Für das Verhalten des Rohres im Fluge und besonders für den Anströmwiderstand ist die Periodizität der Einströmung in das Rohr von wesentlichem Einfluß. Um diese Verhältnisse zu untersuchen, erprobte *P. Schmidt* im Jahre 1944 eine nach besonderen Gesichtspunkten ausgebildete Speicherhaube, die stetiges Einströmen in die Haube bei sicherem Rohrbetrieb ergeben sollte. Für die Versuche verwendete er ein dem Argus-Schmidtrohr der V1 ähnliches Rohr von 100 mm Dmr. und 1 m Länge, das zunächst mit einer Haube nach Bild 6, wie sie bei der V1 verwendet wurde, und dann mit der Speicherhaube, Bild 7, versehen war. Bei diesen Prüfstandversuchen befand sich ein Fadenbüschel in der Nähe des Haubeneinlasses. Man erkennt am Flattern der Fäden das heftige Pulsieren der Strömung an der V1-Haube. Dagegen zeigt Bild 7, daß sich beim Versuch mit der Speicherhaube die Fäden in Ruhe befanden, daß also die Einströmung in diesem Fall stetig war. Das Rohr hatte in beiden Fällen gleiche Leistungswerte.

Die vorstehend erwähnten Möglichkeiten einer weitgehenden Abänderung der Grundform des Schmidtrohres nach Bild 1 ergeben eine vielfältige Anwendbarkeit.

Anwendung

Flugtechnik

Die erste Verwendung in der Flugtechnik war das Argus-Schmidtrohr als Antrieb der V1. Es dürfte von Interesse sein, der bekannten Ausführung der V1 einen Entwurf gegenüberzustellen, den *Schmidt* bereits im Jahre 1934 an die Behörde eingereicht hatte, der aber nicht beachtet wurde, Bild 8 bis 11. Dieser Entwurf zeigt eine wesentlich günstigere Anordnung des Antriebs als bei der V1 und läßt erkennen, daß die flugtechnische Anwendung allgemein größere Bedeutung erlangen kann, vor allem auch für andere Zwecke als für fliegende Bomben.

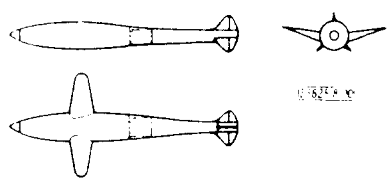


Bild 8 bis 10. Flugtorpedo-Entwurf von 1934.

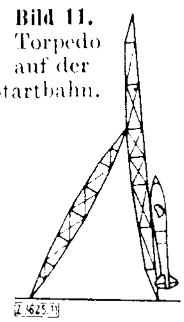


Bild 11. Torpedo auf der Startbahn.

Wegen seiner Einfachheit und Billigkeit eignet sich das Schmidtrohr auch zum Antrieb von Segelflugzeugen. Hierfür wird besonders die in Frankreich entstandene, mit „Escopetto" bezeichnete Ausführungsform vorgeschlagen, während die in den Vereinigten Staaten von Amerika unter dem Namen „Dyna-Jet" hergestellte Bauart zum Betrieb von Flugzeugmodellen dient.

Eine andere, ebenfalls in den Vereinigten Staaten von Amerika eingeführte Anwendung in der Flugtechnik ist der Antrieb der Drehflügel von Hubschraubern[1]). Hierbei werden die Drehflügel durch an den Flügelenden angeordnete Schmidtrohre unmittelbar angetrieben. Der Vorteil besteht in einer wesentlichen Vereinfachung des Triebwerks und im Wegfall des Rückdrehmoments auf die Zelle.

Die Verwendung des Rohres zum Antrieb der Drehflügel von Hubschraubern hat in neuerer Zeit zum Bau eines kleinen Hubschraubers[1]) von nur 91 kg Rüstgewicht geführt [6]. Dabei sind zwei Rohre angeordnet, mit denen ohne Nachtanken 1,5 Stunden bei einem Kraftstoffverbrauch geflogen werden kann, der als nur unwesentlich über dem von Kolbentriebwerken angegeben wird. Das Verhältnis von Rohrgewicht zu Schub beträgt, wie bereits im vorangehenden Hauptabschnitt erwähnt ist, 0,24 kg/kg.

[1]) Bauart der American Helicopter Co.

Heizgeräte und Kohlenstaubbetrieb

Eine kleine Ausführung eines Schmidtrohrs, die als Heizgerät dienen sollte, wurde auf Anregung von *W. Kamm* in den Grundzügen entwickelt[2]) und nach weiterer Ausgestaltung[3]) an anderer Stelle als Stoßbrenner in großen Serien geliefert. Die durch Fortentwicklung nach dem Kriege hieraus entstandene Ausführung ist unter der Bezeichnung Schwingfeuergerät auf den Markt gebracht worden [7]. Wegen der Einfachheit, des niedrigen Gewichts und des unabhängigen Betriebs kann man für die Anwendungen als Heizgerät eine weitgehende Verbreitung erwarten.

Die Unempfindlichkeit des Schmidtrohrs gegen die Brennstoffart ermöglicht z. B. auch den Betrieb mit Kohlenstaub. Erste Versuche, Kohlenstaub bei Einzelzündungen mit Stoßwellen zu verwenden, wurden von *P. Schmidt* schon im Jahre 1932 unternommen, führten aber nicht zum Erfolg. Ein einwandfreier Betrieb mit Braunkohlenstaub wurde erstmalig zu Beginn des Jahres 1945 von *H. Wahl* erzielt.

Im Anschluß an die Versuche von *Wahl* ist es in den letzten Jahren gelungen, den Kohlenstaubbetrieb weiter auszubauen. Die Entwicklungsarbeiten[4]) haben zu einem Betrieb von Schwingrohren mit Braunkohlenstaub und auch Steinkohlenstaub bei beträchtlichen Leistungen geführt. Solche Geräte ergeben einen erheblich größeren Staubdurchsatz als die üblichen Kohlenstaubbrenner, erzeugen zugleich selbsttätig einen Überdruck der Brenngase und erschließen damit neue Möglichkeiten für die stationäre Kohlenstaub-Gasturbine und für andere Gebiete. Ferner gelang es auch[4]), mit Schwingrohren ein brauchbares Schwachgas aus Steinkohlenstaub bei gutem Wirkungsgrad des Gaserzeugers herzustellen.

Bei der Anwendung des Schmidtrohres auf Heizgeräte und Brenner sowie bei der Kohlenstaubvergasung sind für die Leistung ebenfalls die hohen Zündgeschwindigkeiten von besonderer Bedeutung. Die Intensitäten der Zündung und Verbrennung sind so groß, daß beim Kohlenstaubbetrieb Staube von normaler Mahlfeinheit fast vollständig ausbrennen. Bezieht man die Brennerleistung in der üblichen Weise auf das Volum des Brennraums, so ergibt sich bei den bisher als normal geltenden Kohlenstaubbrennern ein Wert von rd. $0{,}5 \cdot 10^6$ kcal je m³ Brennraum und h, bei Wirbelbrennern (oder Wirbelvergasern) rd. $5 \cdot 10^6$ kcal je m³ und h und bei einem Schmidtrohr mittlerer Größe $50 \cdot 10^6$ kcal je m³ und h. Demnach bestehen im Schmidtrohr eine Zündart und Zündwirkung, die noch weit über die Wirkung der starken Strömungsturbulenz in Wirbelkammern hinausgehen.

Schädlingsbekämpfung

Durch die Arbeiten und Erkenntnisse von *K. Stantien* ist dem Schmidtrohr auf dem Gebiet der Schädlingsbekämpfung ein neues Anwendungsgebiet erschlossen worden, das von Bedeutung zu werden verspricht.

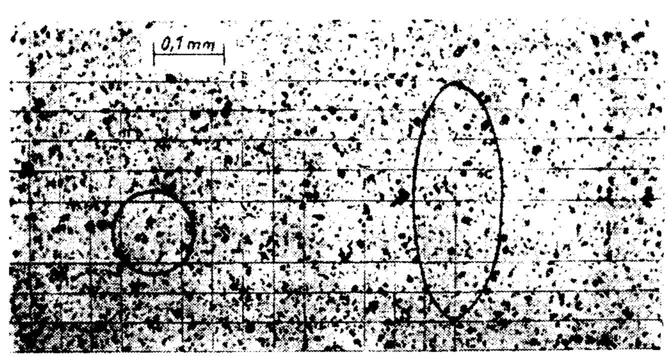


Bild 12. Kondensationsnebel eines Insektizids mit eingezeichneten Umrissen von Schädlingseiern.

[2]) In Gemeinschaftsarbeit zwischen *P. Schmidt* und dem Forschungsinstitut für Kraftfahrzeuge und Fahrzeugmotoren, Stuttgart-Untertürkheim.

[3]) Bauart Kärcher.

[4]) Entwicklungsarbeiten der Ruhrgas AG.

Für die Schädlingsbekämpfung ist es wichtig, daß die Insektizide genügend fein verteilt werden und man eine gleichmäßige Größenverteilung der Teilchen erhält. Die vorausschauende und langjährige Arbeit von *K. Stantien* auf diesem Gebiet hat schon frühzeitig die Möglichkeit einer besonders günstigen Wirkung des pulsierenden Strahls aufgezeigt und bestätigt. Durch das Einführen der temperaturempfindlichen Insektizide nahe dem Rohrende werden die Stoffe fein aufgeteilt und auch genügend schonend behandelt. Hierbei dürfte das pulsierend auftretende Rücksaugen von Atmosphärenluft am Rohrende und die Wirkung der im Rohr laufenden Verdichtungswellen eine besondere Rolle spielen. Einen Eindruck über die Feinheit und Gleichmäßigkeit der Verteilung von Insektiziden gibt Bild 12 (mikroskopische Vergrößerung). Die eingezeichneten Umrisse von Schädlingseiern zeigen die sichere Wirkung der Verteilung und lassen zugleich die sparsame Verwendung des Insektizids infolge der Gleichmäßigkeit der Teilchen erkennen. Eine derartige Gleichmäßigkeit und Feinheit der Verteilung war mit mechanischer Zerstäubung bisher nicht erzielt worden.

Schrifttum

[1] *P. Schmidt*: Und noch einmal V1. Mot.-Rdsch. Bd. 18 (1948) S. 171/72.

[2] *P. Schmidt*: Die Entwicklung der Zündung periodisch arbeitender Strahlgeräte. Z. VDI Bd. 92 (1950) S. 393/99.

[3] Belgisches Patent 502 046 (entspricht deutscher Patentanmeldung Sch 1552 IVb/12g von *P. Schmidt*).

[4] *A. H. Howland* u. *W. A. Simmonds*: A mechanism of intense gaseous combustion. J. Inst. Fuel Bd. 24 (1951) S. 252/56.

[5] *K. v. Gersdorff*: Das Pulsotriebwerk als neuartige Antriebsart für Motorsegler und Sportflugzeuge. Flugwelt Bd. 3 (1951) S. 210/14.

[6] Neue Hubschrauber. Flugwelt Bd. 3 (1951) S. 382.

[7] *L. Huber*: Das Schwingfeuergerät. Automob.-techn. Z. Bd. 53 (1951) S. 209/13; vgl. ferner auch die Zuschrift: Das Schmidtrohr als Heizgerät. Automob.-techn. Z. Bd. 54 (1952) S. 114.

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INTERAVIA VOLUME VIII — N° 6, 1953

The SNECMA Escopette Pulse-Jet

by J. Bertin, in charge of Special Projects, F. Paris and J. Le Foll, Engineers, SNECMA, Paris

I. *Historical Survey*

The SNECMA *Escopette* pulse-jet, which has been fitted in two *Emouchet* gilders built by SEVIMIA (Société d'Etudes Victor Minié), is the first power plant without moving parts—excepting for obvious reasons rockets—which enables a piloted aircraft both to take off and to fly entirely independently.

The history of this engine goes back to the end of 1943. One of the authors, convinced of the value of constant volume combustion (the Lenoir cycle without pre-compression has a theoretical efficiency of 27%), had the idea—which he thought was a new one—of an engine with intermittent combustion and hence with partially constant volume of the burning gases. His original design was roughly as shown in fig. 1, with a valve at the head of the pipe.

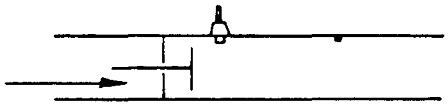


Fig. 1 : Entry valve, controlled by gas pressure, at the forward end of a pulse-jet pipe.

Today, of course, everybody recognizes a diagram of the principle of a pulse-jet with flap valves. But at the time the author knew nothing of this, although he had been working on aircraft power plants for a long time.

The idea was submitted to Chief Engineer R. Marchal, head of the Centre d'Etude des Moteurs à Huile Lourde (Test Centre for Heavy Oil Engines) *, who showed interest in it but raised certain objections to the use of valves.

He feared they would not be sufficiently tight and would create losses due to the inevitable dephasing of the movement of gases in relation to that of the valve. A new suggestion was made, that a spring be placed on the valve with a frequency equal to that of the movement of the gases.

It might thus be possible to reduce the dephasing to a minimum, as the energy taken up from the flow to move the valve is practically negligible.

* C.E.H.L. was an organization under the Technical and Industrial Directorate of the Air Ministry and was at Lyons at the time. Later, in 1947, it was incorporated in SNECMA. Its head, M. Marchal, had meanwhile become Technical Director of this company.

Although this provided a fairly effective solution to one at least of the problems, Mr. Marchal suggested that all moving parts should be eliminated and the valve replaced by a device which favours flow in one direction and hinders it in the opposite direction. This device was named a " detector " (an analogy with the well-known element in early radio sets. Ed.). He suggested the use of vortices as a means of differential throttling. Historical development was then as follows:

— July 10th, 1944, first " vortex detector " was designed,

— the end of 1944 and 1945 were spent in giving this detector satisfactory characteristics,

— May 1946 first correct pulse combustion was obtained, but with auxiliary blower to assist admission,

— 1946 to end of 1947, systematic development of new types of detector,

— April 15th, 1947, first running of a pulse-jet without flap valve yielding static thrust (Engineer Sarrazin),

— March 1950, design of *Escopette* in present form,

— December 19th, 1950, SNECMA's test pilot Gouel took off at Melun-Villaroche in the *Emouchet* glider fitted with four *Escopettes* (installation in collaboration with Engineer Jarlaud of SEVIMIA).

As already mentioned, the author originally believed his idea to be a new one, but was not long in discovering that this was not the case. Technical literature and above all patent specifications showed him that a great many persons had already worked on the problem and that the idea of the pulse-jet, with or without flap valves, was clearly glimpsed by some of them as far back as 1908.

The discovery of a phenomenon on which the functioning of a pulse-jet is founded,

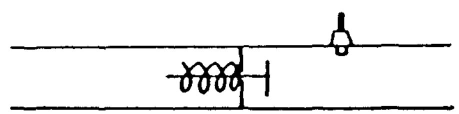


Fig. 2: Spring-loaded entry valve (with frequency adjustment) at the forward end of a pulse-jet pipe.

namely the negative pressure which follows the emptying of a pressurized container when suddenly opened (sometimes called " Kadenacy effect ") may even be said to go back to Huygens, who described a prime mover working on this principle with gun powder.

However, the first models of pulse combustion chambers did not appear until the 20th century. They used the negative pressure effect discovered by Huygens as an accessory to permit of successive fillings of a chamber to provide approximately constant volume combustion.

The important characteristic, it should be stressed, is this phenomenon of constant volume of the burning gases. In this sense pulse-jets are directly related to piston engines and differ from turbojets and ram-jets, in which the gases are burned under more or less constant pressure.

There is another little-known category of reaction engine—piston engines with a jet pipe—in which only the exhaust energy and thermal losses are used for propulsion, the connecting rods being used only to fill the chamber and provide pre-compression. This system, put forward by Lorin in 1908, was seriously examined during the last war but then dropped.

The only way in which pulse-jets differ from the latter type of engine is in their automatic filling, which permits of great simplicity of design. This simplicity has attracted the interest of a great many researchers.

Holzwart, for example, began working on a combustion chamber with automatic valves to feed a gas turbine in 1908. Marconnet, in 1909, suggested models very similar to the *Escopette*; in 1910 Caravodine made a gas turbine actuated by a combustion chamber without pre-compression, with an efficiency that would not be regarded as ridiculous even today. Esnault-Pelterie also made an important contribution to these studies.

Thus the basic principles of intermittent combustion and even the main special equipment, such as flap valve grids, had been worked out before 1914.

After 1914 all this seemed to be forgotten and no further traces of work in this field were found. It was not until 1930 that the subject was taken up again in various countries, and only the work carried out in secret in Germany by Paul Schmidt produced a practical model, namely the Argus-Schmidt tube used in the V 1 flying bomb.

This engine was not known in France until the beginning of 1945. The work described in this article had begun before that date, and in any case the Argus-Schmidt tube had mechanical flap valves, whereas the SNECMA pulse-jet eliminated these from the beginning.

It is this new aspect of the question which is dealt with in greatest detail in the following paragraphs.

II. General remarks on functioning

As the general functioning of the *Escopette*, apart from the detector, does not differ to any marked extent from German pulse-jets, we shall do no more than quote a passage from a memorandum by Paul Schmidt which adequately describes its kinematics.

" The figure (Fig. 3) shows a diagram of the kinematics of a pulse-jet with its different operating phases.

" The pipe has an admission valve for the combustion air (left in the diagram, indicated by a double line). This permits the entry of a flow of fresh air enriched with petrol, but prevents the combustion gases from returning after ignition of the mixture, i.e., when there is positive pressure inside the pipe.

" The top diagram (a) shows conditions at the beginning of the cycle : there is a charge of fresh mixture near the admission valve. The rest of the pipe contains burnt gases from the previous combustion and even fresh air near the exit orifice.

" Diagram b shows the position immediately after ignition, with three small arrows indicating the effect of the pressure due to combustion.

" This pressure advances very rapidly, and the column of gases in the pipe is pushed towards the rear (diagrams c to f), a movement which continues even after the positive pressure has disappeared, so that air is drawn in through the admission valve. The negative pressure now in the pipe, particularly near the valve, draws the column of gases back into the pipe, as indicated by the rear arrows in diagrams g, h and a. Because of this return movement, air also enters the pipe through the exhaust orifice but will be expelled again during the next cycle.

" Thus the pipe works on a definite frequency. In addition to the automatic admission of fresh air, there is automatic ignition at a high combustion speed."

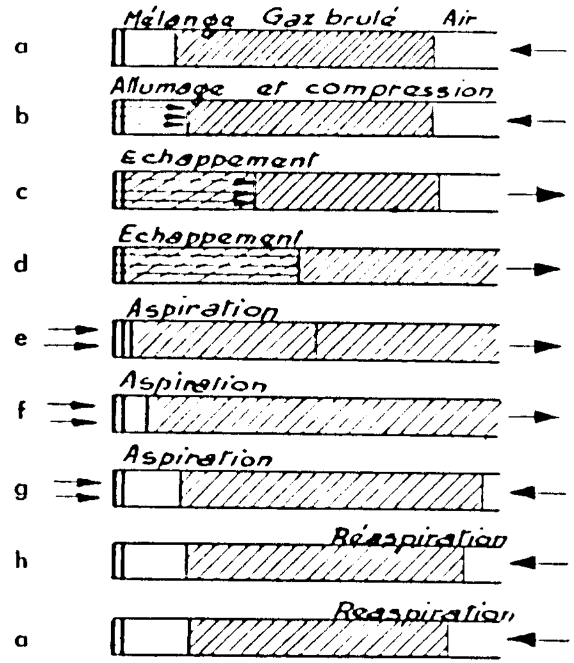


Fig. 3 : Kinematics of a pulse-jet according to Paul Schmidt. a—mixing of burnt gases with air ; b—ignition and compression ; c—exhaust (phase I) ; d—exhaust (phase II) ; e—suction (phase I) ; f—suction (phase II) ; g—suction (phase III) ; h—return suction (phase I) ; a—return suction (see above).

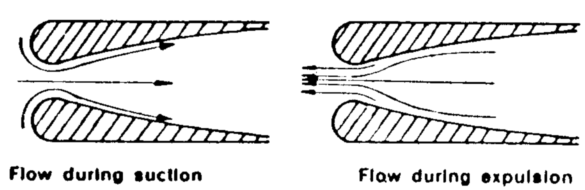


Fig. 4 : Venturi tube as simplest example of a " gaz flow detector ".

It will be seen that the column of gases in the pipe plays a double role in the combustion process :—

a) during combustion it resists the expansion of the burning gases, thus putting them under positive pressure, an essential condition for the production of motive power ;

b) after combustion it furnishes the work required for the admission of a new charge of air-fuel mixture.

Fig. 5 : Flow through an entry detector during suction (unbroken line) and during expulsion (broken line) as a function of the pressure ratio.

III. *Detectors*

These essential functions can still be obtained if the flap valves are replaced by an orifice which is more permeable in the admission direction than in the return direction. The ratio of permeability (in the two directions), or " coefficient of detection ", must be sufficiently large. Such an orifice is known as a " detector ". Its simplest example is a venturi tube (fig. 4).

The Mach number at the neck, in both directions, depends solely on the ratio of the upstream pressure to the pressure at

the neck (or downstream). The same Mach number can be obtained at the neck with a smaller ratio between upstream and downstream pressures in the admission direction than in the opposite direction.

The detectors were the objects of a large number of experiments which included the measurement of flow as a function of the ratio between upstream and downstream pressures in the two directions (fig. 5). During these experiments attempts were made to meet the following conditions :

— high flow in the admission direction
— high permeability ratio
— light weight and ease of construction
— smallest possible volume.

Two types in particular were examined: the vortex detector (fig. 6) and the funnel detector (fig. 7). Both of these had high permeability ratios (5 and 4 respectively).

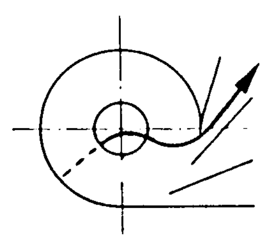


Fig. 6 : " Vortex " detector : the arrow marks the direction of suction.

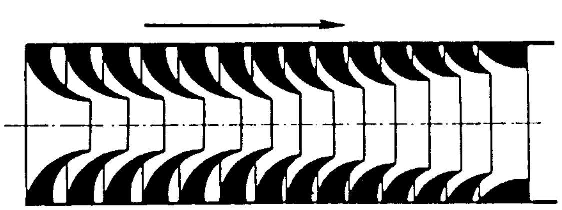


Fig. 7 : " Funnel " detector : the arrow marks the direction of suction.

The latter model was found to be better than the former, in spite of its lower permeability ratio. This unexpected result can be explained by examining the dynamic aspects of " variable flow ".

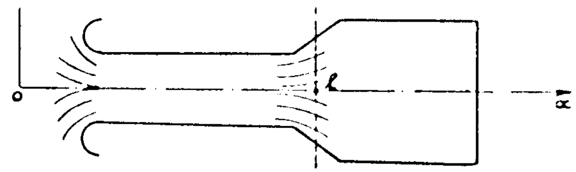


Fig. 8 : General arrangement for a variable flow : pressure (p) and velocity of flow (v) are functions of time (t) and position (x) ; allowance also has to be made for density ρ.

IV. *Variable flow*

Let us imagine a container connected to the atmosphere by a cylindrical pipe with a rounded lip, and assume (fig. 8) that :

— the fluid is incompressible in the pipe, but compressible in the container,
— the flow is unidimensional and adiabatic in the pipe, its lip to the atmosphere and transition part to the container,
— the container has previously been placed under negative pressure.

Let initial time t_1 be that at which atmospheric pressure is reached again in the container and let v_1 be the speed in the pipe at this moment.

Euler's equation is written :

$$\frac{\partial p}{\partial x} + \rho\left(v\frac{\partial v}{\partial x} + \frac{\partial v}{\partial t}\right) = 0$$

and the equation of continuity in the cylindrical pipe :

$$\rho\frac{\partial v}{\partial x} = 0$$

v and $\partial v/\partial t$ are functions of t, so that Euler's equation can be integrated as follows :

$$p\,(l,t) - p\,(0,t) + \rho l\frac{\partial v}{\partial t} = 0$$

As the atmospheric pressure can be regarded as applying to point of origin 0, or : $p\,(0,t) = 0$ this equation can be integrated between t_1 and t_2, thus giving the areas hatched in fig. 9 :

$$P\,(t_2) = \int_{t_1}^{t_2} p\,(l,t)\,dt$$
$$P\,(t_2) = \rho l\,(v_1 - v_2)$$

Numerical application :

$$\rho = 10^{-3}\ t \cdot m^{-3}\ ;\ l = 0.5\ \text{m}$$
$$v_1 = -v_2 = 200\ \text{m/sec}$$
$$P\,(t_2) = 10^{-3} \times 0.5 \times 400 = 0.2\ \text{pz} \cdot \text{sec}$$

For example, for $t_2 - t_1 = 2 \times 10^{-3}$ secs, mean positive pressure equals 100 pz, which corresponds to a maximum pressure of 120 to 150 pz for the habitual forms of pressure distribution versus time.

Once p, l, v_1 and v_2 are fixed, area $P\,(t_2)$ is determined. A combustion or for example a supply of fluid through a second orifice (not shown in fig. 8) act of course on the mean positive pressure.

The numerical values envisaged show that it is possible to maintain a marked positive

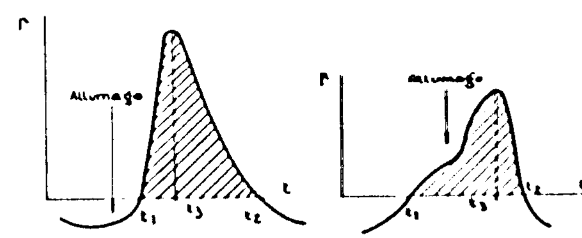


Fig. 9 : Pressure-time diagrams for early ignition (left) and late ignition (right).

pressure throughout combustion (2 to 3 milliseconds) by means of the processes which

have just been described. The compressibility of the gases modifies this picture and makes it more complicated, particularly when the pipe occupies a large proportion of the total volume. However, it gives a fairly accurate idea even when the flow is transonic (cf. fig. 12).

*

V. *Application to the pulse-jet*

Paul Schmidt placed considerable emphasis on the fact that the negative pressure which fills the combustion chamber from the front also produces a return movement by the gases contained in the exhaust pipe, a movement which in turn hinders the rearward expansion of the gases during combustion, or even creates a slight pre-compression. Applied to the exhaust pipe, the preceding calculations illustrate Paul Schmidt's affirmation and make it easier to understand the thermodynamic principles of pulse-jets with flap valves.

Thus the movement of the column of gases resists expansion of the burning gases and creates a positive pressure whose mean value is inversely proportional to the duration of combustion. This shows the advantage of short-period combustion. In fig. 9 it will be seen that between t_1 and t_2, the specific volume of the gases in the container decreases at first up to a certain moment t_3, then increases.

The optimum moment for ignition depends on the relative increase in temperature caused by the combustion, the duration of the latter and the form of pressure distribution versus time, but it is always before t_3. If combustion is very slow or produces a substantial increase in temperature (complete filling and high richness), this moment may be earlier than t_1. In actual fact, filling is always very incomplete, which greatly reduces the relative increases in temperature and pressures produced by the combustion. These major disadvantages are balanced by a high speed of combustion. In general the ignition point is between t_1 and t_3, and fairly near to t_3.

The real cycle is therefore more closely related to that of Beau de Rochas than that of Lenoir, but the dilution of the fuel mixture in the burnt gases limits the theoretical efficiency to a value very much lower than had been expected.

VI. *Application to the detector*

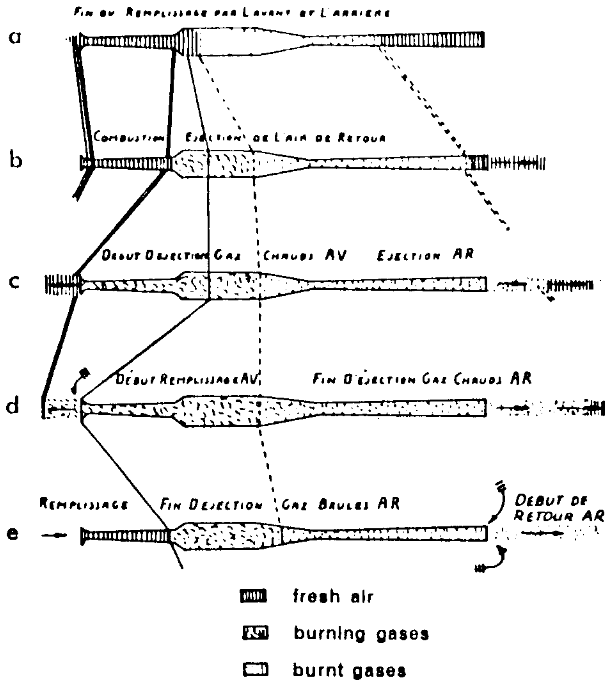
The above examination shows that once launched towards the chamber, a column of fresh air can " close " the latter during combustion. We can now turn to the admission, applying this phenomenon to the front of the pipe, combining it with the detection pure and simple described earlier. The only pulse-jets without flap valves which have ever functioned with performance comparable to that of Schmidt's tube used this last principle. More exactly, the favourable inertia effect shows itself spontaneously as soon as the column extends up the chamber over an appreciable length. This is what happened with the device known as the " funnel detector " and explains its superiority over the " vortex detector ". Later we made systematic use of the effect of inertia, sacrificing detection proper in the " smooth-tube detector " which is an extended venturi tube (see fig. 12).

In the preceding we have above all examined the effect of inertia which tends to compress the gases in the chamber ; but there is also an inertia effect in the other direction which tends to empty it. It is this effect, in the exhaust pipe, which enables the chamber to be swept out, the admission being made via the other orifice. In the detector, on the other hand, its effect is to increase escape forwards and to hamper filling. But this escape does not occur until after combustion, and thus has no direct detrimental effect on thermal efficiency. Moreover the burnt gases leaving the chamber cannot be ejected until they have pushed before them a large volume of air that has not been mixed with fuel. The movement stops in general at the beginning of their final ejection (fig. 10c and 10d).

The forward-moving burnt gases, which no longer contain oxygen, are again mixed with fuel as they pass the injector zone, but the fuel is not lost, since they are almost entirely reabsorbed during the next sequence (fig. 10e). The fuel is even completely vaporized, heated and thoroughly mixed, which assists rapid combustion. A rough atomization of the fuel, and hence a very moderate injection pressure, are therefore sufficient. On the other hand, there is a reduction in filling. Figs. 10, 11 and 12 show the different phases of operation. They were produced by Mr. Salmon on the basis of pressure and temperature measurements made at several points of the pulse-jet by means of recording instruments provided by SNECMA's measurements services under M. Fleury. The values of pressure (p) absolute temperature (T) and $\partial p/\partial x$ having thus been measured, it was possible to calculate speeds and determine trajectories by integrating Euler's equation :

$$\frac{\partial p}{\partial x} = \frac{-1}{R}\,\frac{p}{T}\,\frac{dv}{dt}$$

The ejection of air forwards at a high speed would result in a considerable loss in



$t = t_a$ — *end of filling from front and rear.*
$t = t_b$ — *combustion and rearward expulsion of return suction air.*
$t = t_c$ — *beginning of exhaust of hot gases forward ; expulsion rearwards.*
$t = t_d$ — *beginning of filling from front ; end of expulsion of hot gases rearwards.*
$t = t_e$ — *filling ; beginning of return suction.*

Fig. 10 : Diagram of operation of a pulse-jet with smooth-pipe detector.

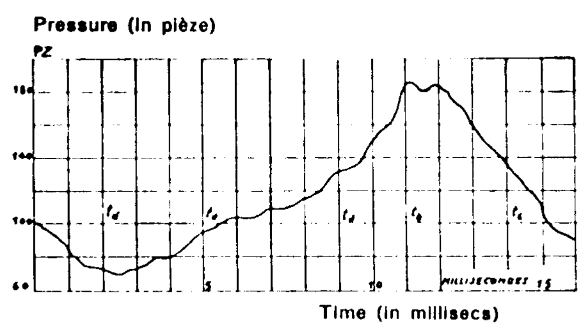


Fig. 11 : Pressure-time diagram for pulse-jet with smooth-pipe detector ; the phases shown in fig. 10 are marked on the abscissa.

thrust. It is therefore necessary to use a device to turn the air jet backwards without disturbing the admission too much.

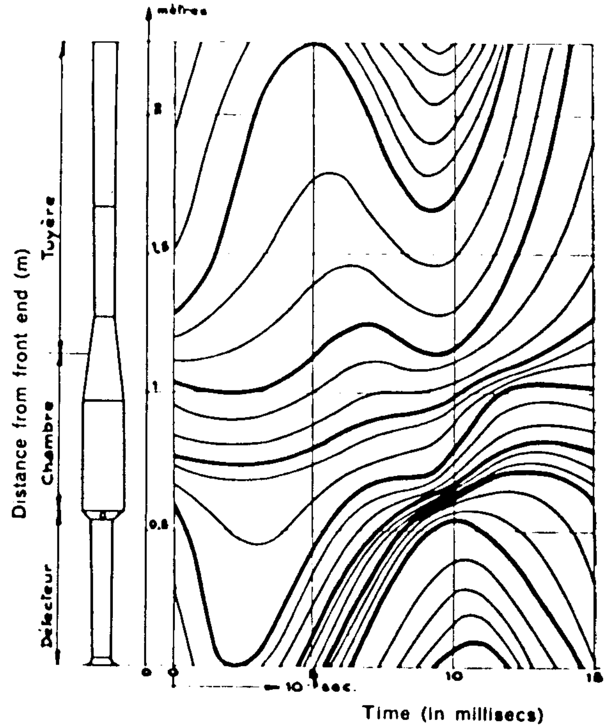


Fig. 12 : Gas movements in pulse-jet with smooth-pipe detector, as a function of time (in milliseconds).

Making allowance for the small bulk of this device, it was possible to obtain specific thrusts (in relation to either weight or frontal area) superior to those of the V 1 power plant. Specific consumptions are also noticeably lower.

VII. *Life of pulse-jet without flap valves*

The short life of the V 1's power plant was due solely to the low endurance of the flap valves, so that if relatively cold detectors with no moving parts are used, the engine's life then becomes that of the hot part. The elimination of mechanical flaps also does away with shocks and their harmful effects.

By including a simple spring in the engine suspension it is possible to eliminate all vibrations that would otherwise be transmitted to the airframe. Simultaneously the alternating stresses inside the engine itself are reduced to the minimum compatible with the periodic nature of its operation.

The only major stresses that remain are those that result from the variations in internal pressure, as the pipe is alternately under positive and negative pressure. The pipe is circular in section.—Although the fatigue strength of stainless steel decreases rapidly with temperature, the endurance of the pipes was found to be definitely superior to what had at first been expected, despite temperatures of the order of 850° C.

The results of the experiments were encouraging (fig. 13), thanks to improvements in manufacture, particularly a careful choice of the position of the welding seams, the utmost attention to the welding itself and a strict inspection of the metal sheets used. Ruptures occur as a rule independently of the seams, after a certain time which varies according to the nature and thickness of the material used. For example, the sheets of stainless steel with low nickel content used in the *Escopette* were 4/10 to 6/10 mm thick depending on where they were used, and a reduction of 10 % in one of these thicknesses would reduce the endurance from about 50 hours to 5 hours.

To sum up, the very short life of pulse-jets with flap valves could only be lengthened to several hours at the expense of the thrust, and the engine would be suitable only for use in an expendable machine (the V 1 for example). On the other hand the pipe of a pulse-jet without flap valves enables it to be used as a " consumable " engine or even a long-life engine, depending on the weight selected. Even now our pulse-jet, with a life of about fifty hours, has a thrust/weight ratio of about 6.

VIII. *Glider with auxiliary pulse-jets*

The basic thrust of the pulse-jet with a

single detector is rather low, and the diameter of the pipe is small.

However, several pulse-jets of this type (four and six *Escopettes*) have been used experimentally as an auxiliary power plant in SEVIMIA *Emouchet* gliders, to provide practical experience. There is no need to go into the value in gliding of a device of this kind to provide take-off power and raise the glider to an altitude of 3000 to 5000 ft. if required. The glider is then relieved of all the limitations imposed by the use of winches or towing aircraft. Furthermore, the *Escopette* pulse-jet was designed with particular regard to its utilization at low speeds where its output and efficiency are very good, consumption being only 1.8 lb./lb.t./hr, or about twice the consumption of a good turbojet. Its simplicity of construction gives it other advantages, such as strength, light weight, low cost. Then there is the key advantage of these engines, namely low fuel injection pressures : 1 to 2 atmospheres at the most.

The *Escopettes* were mounted in two *Emouchet* SA.104 gliders belonging to the Sports and Light Flying Service, in collaboration with Engineer Jarlaud. The tests proved the value of the arrangement, and in particular the fact that the performance of the glider is relatively little affected by the presence of the stationary engines, at any rate in a glider of moderate lift/drag ratio, i.e., not designed for high performance (fig. 14). Better results would probably be obtained with an airframe designed from the outset to take pulse-jets as an auxiliary power plant. The life of the airframe does not appear to have been affected by the vibrations, as had been feared. The simple spring in the engine suspension proved its worth.

Several expert pilots, including Eric Nessler, the well-known champion, have flown these machines with full satisfaction. In addition to the simplified take-off, there was another advantage which they particularly appreciated, that of being able to re-start the engines at any time during flight. This is possible thanks to the air flow. All that is necessary is to inject the fuel and provide a short series of sparks in the plug. This, for example, would enable a glider school to avoid landings in the open country and the expenses these entail.

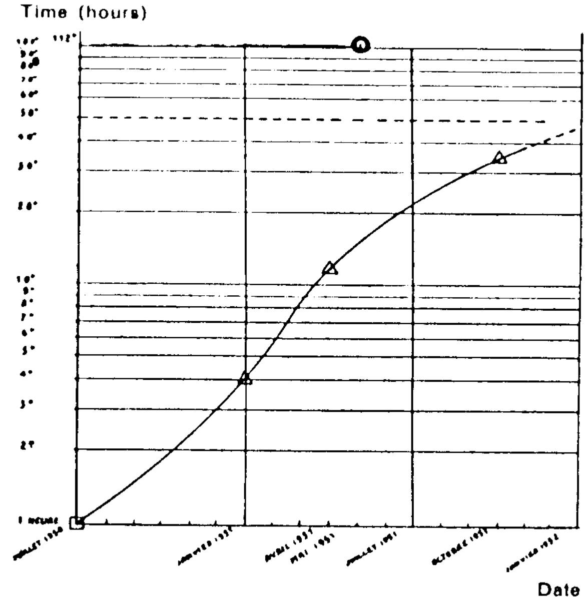


Fig. 13 : Diagram of tests on SNECMA Escopette.

□ *official one hour test run on bench*
△ *average running time during practical tests in the* Emouchet *glider*
○ *bench tests under simulated flying conditions (3 mins. idling plus 10 mins. max. thrust)*
--- *anticipated minimum endurance*

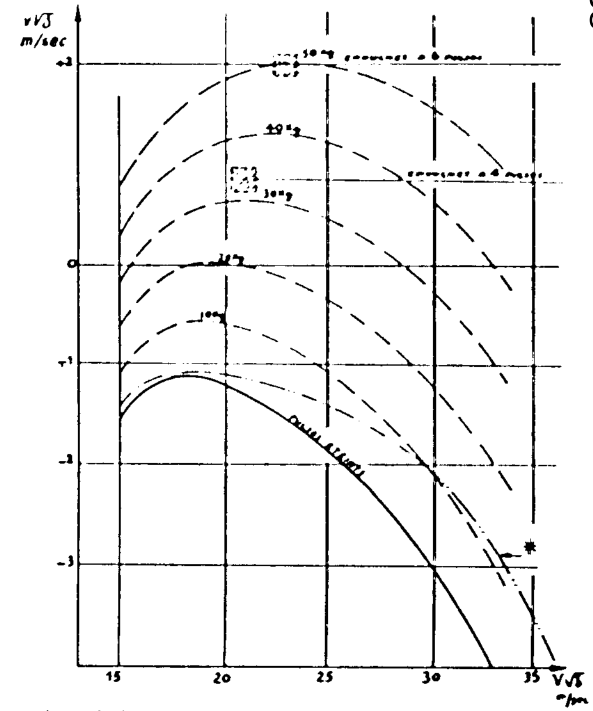


Fig. 14 : Polars of the Emouchet-Escopette *(gross weight 325 kg) for various thrusts of its pulse-jets (four or six with max. thrust of 10 kg. each). Polar of the glider without power plant is shown by dot-dash line (gross weight also 325 kg.).*

IX. *Conclusion*

The fitting of pulse-jets to gliders is not the sole object of these experiments, which are rather aimed at making the pulse-jet into a real aircraft power plant capable of use for a variety of purposes. Considerable increases in power and efficiency have already been obtained compared with the *Escopette* fitted in the *Emouchet*, yet the margin between the present achievement and the theoretical limits is still great enough to give ample scope for improvement (200 to 300 %).

This gives us great hopes for the future of the pulse-jet, as no other form of power plant now available has such a wide development potential.

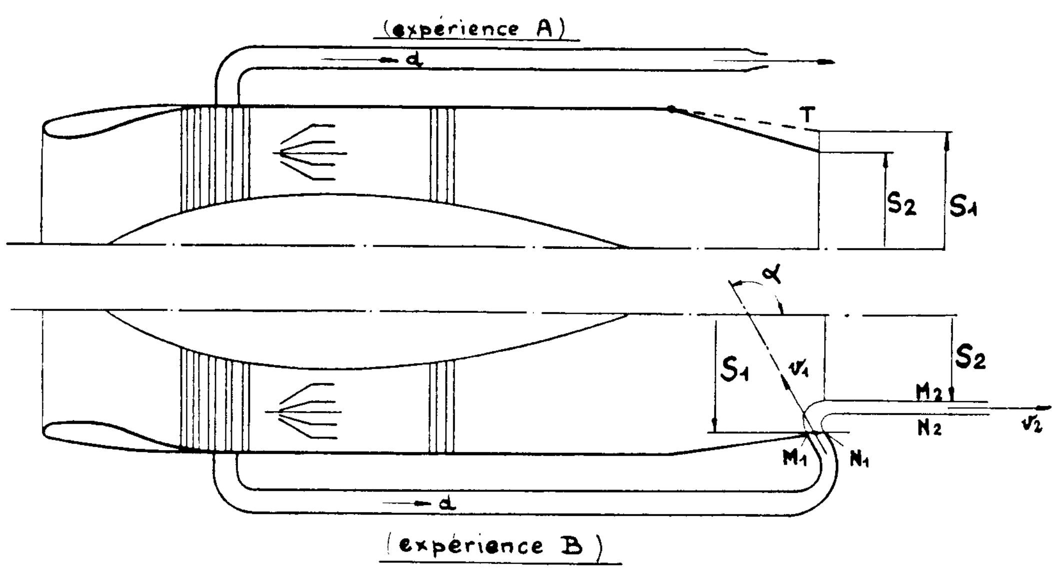
COMPTES RENDUS
ACADÉMIE DES SCIENCES. 8-22-55 Vol. 241

AÉROTECHNIQUE. — *Contrôle par jet transversal de la section d'éjection des tuyères à réaction.* Note de MM. **François Maunoury**, **Marcel Kadosch** et **Jean Bertin**, présentée par M. Maurice Roy.

Pour adapter les réacteurs d'aviation aux divers fonctionnements, en particulier dans l'utilisation de la rechauffe, la tuyère d'éjection à section variable s'est largement répandue, sa section de sortie pouvant varier du simple au double.

Nous allons exposer diverses formules simples appliquées à ce problème, dont la vérification expérimentale a été obtenue au cours de la mise au point du procédé étudié dès 1950 par la Société d'Étude et de Construction de moteurs d'aviation et adopté récemment sur une série de réacteurs construits par cette Société.

Le procédé utilisé consiste à diriger sur le flux à contrôler un jet auxiliaire de gaz, à pression totale supérieure à la pression statique de l'écoulement, le jet auxiliaire abordant ledit écoulement sous une incidence appropriée. Dans l'application ici envisagée, le jet auxiliaire provient d'un prélèvement effectué en amont sur le flux principal, en général à partir du compresseur ou des chambres de combustion ; mais cela n'est nullement une nécessité et l'on peut dans certains cas utiliser un générateur séparé.



Considérons un réacteur sur lequel un débit-masse réduit d a été prélevé, le débit principal résiduel D étant évacué par la tuyère d'éjection T (*fig.* 1).

Dans une expérience A, le débit prélevé d est détendu dans une tuyère t et éjecté comme D, tandis que la tuyère T, variable mécaniquement, peut réduire de S_1 à S_2 la section d'éjection de D.

Dans une expérience B, le débit prélevé d est détendu et injecté dans la tuyère T, fixe et de section de sortie S_1, par une fente périphérique de section s_1, avec une inclinaison α sur l'axe de T, les conditions au prélèvement et en amont de l'injection étant les mêmes que dans l'expérience A.

On notera que si, dans cette seconde expérience, la conjonction des flux D et d s'opère en subsonique et si leur mélange, après l'injection du second, ramène celui-ci sans perte sensible d'énergie à une direction parallèle à l'axe de T après détente complète, la variation totale de quantité de mouvement suivant l'axe de T, donc aussi la poussée, reste la même dans les expériences A et B.

Supposons l'écoulement uniforme dans les sections d'indices 1 et 2, et notons p_2 la pression, admise constante le long de la ligne de jet $N_1 N_2$. Par application du théorème des quantités de mouvement et en fonction des nombres de Mach M_1 et M_2, on obtient : — pour expression du coefficient de striction

$$\sigma = \frac{S_2}{S_1} = \frac{M_1}{M_2}\left(\frac{1+\frac{k-1}{2}M_2^2}{1+\frac{k-1}{2}M_1^2}\right)^{\frac{k+1}{2(k-1)}}$$

— pour la quantité de mouvement finale de la veine principale :

$$\frac{DV_2}{S_1 p_2} = kM_2^2\sigma$$

— et pour la variation de quantité de mouvement du jet auxiliaire entre M_1N_1 et M_2N_2 selon la formule due à MM. L. Bauger et P. Gélin :

$$\frac{dv_2 - \cos\alpha(s_1p'_1 + dv_1)}{S_1p_2} = (1 + kM_1^2)\left(\frac{1+\frac{k-1}{2}M_2^2}{1+\frac{k-1}{2}M_1^2}\right)^{\frac{k}{k-1}} - (1 + kM_2^2\sigma)$$

en désignant par p'_1 la pression moyenne du jet auxiliaire en M_1N_1, et par v_1, v_2, les vitesses moyennes du même jet en M_1N_1 et M_2N_2 respectivement.

Le minimum théorique de σ correspond à $\alpha = 180$ degrés. Mais, comme il peut être pressenti, l'expérience montre que le jet auxiliaire ainsi injecté tangentiellement et à contre-courant se perd en partie dans la couche limite vers l'amont avec un mauvais rendement par dissipation d'énergie et que son retournement vers l'aval occasionne une nouvelle et importante dissipation d'énergie. Il résulte d'essais systématiques que, pour une application du genre ci-dessus, le minimum réel de σ correspond à $\alpha = 140°$ environ.

Il doit être signalé que nos expériences ont fourni une excellente vérification des formules ci-dessus. En particulier si pour $\alpha = 90°$, on augmente v_1 et v_2 (par exemple, en élevant la température du flux auxiliaire), le débit de prélèvement d varie, à σ constant, en raison inverse.

BRIEF HISTORY:

The pulsejet engine was developed in the late 1930s by the Germans for use in their (in?)famous V2 rockets, otherwise known as "buzzbombs."

The unique engine in these crudely guided missiles propelled the rocket to somewhere over its intended target, where it promptly turned itself off, and then glided for several seconds in free fall/flight. When the rocket hit the ground, it detonated, causing mass death. Part of the reason they were so feared is precisely because they were not very accurate, nor was it known in the final seconds before detonation where the rocket was going to hit. Those few seconds of silence before the explosion led to increased suspense in the victims of its attack.

FUNCTION:

The engine works in the following manner:

Air is forced through several one way valves via the intake manifold on the front of the engine. It is mixed with atomized gasoline or other fuel either before or after it passes through these valves, and then enters the combustion chamber. The air/fuel mixture is then ignited with a heat source (spark). The resulting explosion forces its way out the back of the combustion chamber, through the exhaust manifold, resulting in a Newtonian-second-law-reaction, thereby pushing the rocket forward. These expanding gases are not allowed to exit the front because the intake valves only allow entrance to the chamber.

After the explosion there is a volume of low pressure inside the combustion chamber. More air and gas is drawn in through the front valves as the pressure finds equilibrium with the atmosphere, as well as in through the exhaust manifold. The air and gas mixture is again ignited, and the cycle repeats itself.

THE MODEL:

The plans available through *Jet Technologies* are a straightforward, simple rendition of this type of engine. It can be built from materials found at home or, mostly, in the hardware store. According to them, this model will produce 7 pounds of thrust and propel a model plane up to 200mph. This engine burns normal gasoline. Should you choose to use this engine in a model airplane, be warned that the finished version is approximately 4.5 to 5 feet long, including the tail pipe, and weighs several pounds. Proper heating insulation is required so that the plane model itself does not burn up from coming in contact with the combustion chamber. Because a model is meant to be taken to a flying field, certain modifications and/or additions have been added at the end of this article in order to make building and transporting the ignition system (ground based only), and on board fuel delivery system, more portable. These are the author's own ideas and are not in any way affiliated with *Jet Engine Technologies*.

Combustion Chamber:

The majority of the air drawn into the combustion chamber after ignition comes in through the tail pipe. The hot gases from the previous explosion reentering the combustion chamber provide ignition for the next cycle. Obviously, however, the net amount of gases escaping the tail pipe is greater than zero, or else the engine would go nowhere.

The main combustion chamber is a *large* size aerosol can. I used a Niagara Spray Starch can. The bottom is cut out with a can opener. The removal of the top part is a bit more difficult. Pliers and a screwdriver are used to pry open the "lip" that seals the spraying mechanism from the rest of the can. All that is left, then, after the top and bottom are removed is the outside of the can and the semi-circular section on top of the can that is still attached. Please see the accompanying illustration. A hole is drilled 3 inches or so from the wide (front) end into the side of the combustion chamber so that a spark plug and its accompanying nut can be firmly bolted on. It may be sealed if necessary with high temperature gasket material. The spark plug is screwed firmly into place. Use the smallest plug possible.

Valve Assembly:

The next major feature of the pulsejet engine is the construction of the "valve plate", or reed plate, and the reed valves. These are the one-way valves that allow the mixture of air and gas into the combustion chamber. The valves themselves are made from .005 inch thick feeler gauge, used by auto mechanics. I ordered mine from an auto supply store for about $2-$3 each. They are about 12 inches long, about 3 /8 inches wide, and very flexible. Scissors are then used to cut them to their proper shapes. The best way to describe this shape is the outline of a child-drawn house. Basically a vertical rectangle with a tall triangle ("roof") added on to one end. The "roof" triangle part of the valve takes up approximately one half of the total length. Eight of these are made and they are about an inch long overall.

The valve plate houses this assembly of eight valves, and is made nominally from 1/8" thick aluminum plate. If 1/8" plate is not available, two 1/16" plates which have metal-epoxied together can be used. These are then cut to size (4.5 inches x 4.5 inches). I used a Black and Decker wood saw (which is supposed to be a no-no, so no responsibility is taken or

assumed for any problems that may arise from using it in the following manner) fitted with an abrasive/metal cutting blade. It made a lot of noise, but managed to successfully cut the aluminum.

There is a total of two other plates in the assembly, also made from the same sized aluminum. In addition, there is 1 square of 1/16" high temperature gasket material with the same length and width dimensions as these plates. These 3 plates and a gasket are stacked together and four holes are drilled in the corners. These holes are eventually used to hold the entire assembly together. Unless extreme care is taken or proper tools are available, the plates will most likely come out uneven and not completely square. When this is the case, proper caution must be taken in order to identify top, bottom, left, right, and front and back of each plate, and THESE MUST BE KEPT IN THE SAME ORDER. Otherwise the engine's alignment will be untrue, and it may not work properly. The four corner holes are eventually used to hold then entire engine assembly together, and these need to be aligned properly.

After all three plates and gasket are drilled, the valve plate itself has more holes drilled into it. These are the intake holes and are covered by the valves. They are used to channel the air and gas mixture into the combustion chamber: There is a 1/4" hole drilled dead in the center of the plate. There are eight 5/16" holes drilled evenly around the center, about 1" away. And finally there are another eight holes, a bit smaller than the first set of eight, drilled directly in line between the first eight holes and the center hole. The pattern can best be describe as 8 sets of 2 holes each, arranged radially about the center of the plate, with the large holes at the outer edge of the pattern. In other words, the hole size is proportional to the distance from the center.

To review:

Distance from center of plate (and 1/4 inch center hole) to:

Four corner holes:	1-3./4". (this is new information -- you didn't miss it the first time).	
		3/16" diameter
Outer eight holes:	1-1/16". They have a	5/16" diameter
Inner eight holes:	3/4 ". They have a	13/64 inch diameter

After all the holes are drilled any burrs must be smoothed/ removed, using a file or wire-brush bit mounted on a drill. After the smoothest side of the valve plate is identified, it is covered with a thin coat of ***high temperature silicon sealer***. A copper and/or petroleum based gasket maker MUST NOT be used. Though it may be cheaper than the silicon version, it makes nothing but a big, useless mess, and doesn't cover anything up. Should you unwisely use this type of sealer, you may need to make new valve plates, as I unfortunately discovered the hard way.

After a thin layer of silicon sealer is placed over the valve plate, it is smoothed over with a plastic zip-type sandwich bag before it dries. The point here is to make a *smooth* surface so that the valves can make an air tight seal with the plate. The plans suggest using the edge of a credit card over the plastic bag so that the surface is bump-less and perfectly flat. The type of silicon sealer I used was red in color, smelled bad, and cost around $5.00 for a tube.

The next step is to place a 1/4" bolt through the center hole to hold the valves in place on the silicon side of the valve plate. The bolt presses on the curved cut-out bottom of a soda can to squeeze the valves in the center of the plate, and hold them in place over their respective holes. The valves are arranged radially, like the holes themselves, with the wide end away from the center. By my experience this does not work because there isn't enough force available from the can bottom to keep the valves from sliding around and covering their respective intake holes. Valve adjustment is the most crucial part of building the pulsejet engine because if the valves don't seal properly, there will be "blowback" which will lead to a fire. Instead of an aluminum can bottom, which is annoying if nothing else to cut out, the bolt can be placed through a 3/4" outer diameter washer to hold the valves in place. The bolt is inserted through the valve/silicon side of the plat towards the front, and firmly hand tightened with a proper nut. The bolt must be long enough so that another nut can be fit over it.

Each valve needs to be tested individually: This task is accomplished by putting your mouth up to each hole from the opposite side of the valve (non silicon side), and sucking. There should be a "flip" sound, as the valve is sucked into the silicon surface, after which there should be no air flow. A little leakage on each of the valves quickly adds up to lot of leakage on the entire front plate, so be sure to adjust these valves as leak proof as possible. .A possible suggestion is to flip the faulty ones over and use the other side against the silicon sealer. Make new ones as necessary.

The next plate to make is the intake plate, which is the second of the aforementioned drilled aluminum plates. A large hole is cut in the middle of both the intake plate and the 1/16" gasket, approximately 3 inches in diameter, or enough to go around all the holes in the valve plate. The hole in the intake plate is cut by drilling many smaller holes around a 3" diameter circle, just on the inside of the circumference, using a 1/4 inch bit or smaller. The rough edges are then smoothed out using either a file or an appropriate (metal routing) drill bit. The hole in the high temperature gasket is cut with scissors. Seal this gasket to the plate with silicon sealer.

The inner intake cone is a cone that is screwed on over the remaining part of the center bolt that holds the valves onto the valve plate. It is made from steel soup can metal. Its function is to disperse the air and gas mixture into the intake holes. The cone's outer diameter is obviously small enough so that it doesn't cover up the intake holes. It is about 3 inches tall from tip to base. Instead of riveting or epoxying the holes like it says in the plans, a propane brazing torch and appropriate solder can be used to hold the cone together. A 1/4" nut is then epoxied to the inside of this cone, so that it can fit over the center valve-holding bolt. After the epoxy cures, the inner intake cone assembly is screwed onto the center bolt, before the intake plate and the valve plate are screwed together (in a little while).

There is one more cone that is made in the same manner as the first, and that is the outer intake cone. This cone is larger, such that the wide end fits snugly along the inside edge of the 3" hole in the intake plate. It is metal-epoxied to the intake plate, so that there are no leaks. The cone is also sealed to the intake plate before the intake plate and the valve plate are screwed together. The final product should look just like a flat plate and a cone sticking out of it, with no indication that there ever was a 3" hole in the plate. The cone itself is made as if the point were cut off, so that the hole in front is about 1" - 2" in diameter -- it looks like a miniature traffic cone, It is also about 3 inches tall, from base (primary apex) to top (not tip, the secondary apex; in a normal cone the secondary apex's diameter is zero; in this one it is 1" - 2"). Be sure to trim smoothly the edge of the outer intake cone which may come out the back, through the intake plate. Use tin snips or something.

Up to now, this is what has been put together, going from front to back:

An inner intake cone-- screwed onto a bolt that goes from back to front, which also holds the reed valves onto the valve plate.

An outer intake cone, sealed along the inside edge of the intake plate. Looks like a traffic cone, with no point.

An intake plate -- same dimensions as the reed plate, and with a large hole the middle about 3 inches in diameter. This hole has the outer intake cone in it.

A 1/16 inch thick high temperature gasket -- with a hole the same size and in the same position as in the intake plate.

A valve plate -- with the no-silicon side forward, and which has the inner intake cone attached to it on this (front) side. On the back side of the valve plate is a thin layer of silicon sealer There are a total of 21 holes drilled in this plate -- four corner holes which will eventually be used to hold the entire assembly together; one center hole, through which is the center bolt which holds on the valves, and also to which is attached the inner intake cone; eight small holes drilled about the center; and eight larger holes drilled outside the smaller holes.

Reed valves -- a total of eight, held on by a washer and a central bolt, arranged radially about the center.

Exhaust Pipe Assembly:

The exhaust pipe is made from 1-3/8" outside diameter steel fence post. It is initially cut to 31 inches, using whatever method possible. Wear goggles. There are four holes drilled 90^{o} apart (evenly) into this pipe about 2 inches from the flattest end. (Flat end means the end whose [geometric/circular] plane is nearest perpendicular to the major axis of the pipe.) Into these holes are put small sheet metal screws which will keep the exhaust plate from coming too far forward, and help to hold everything tight. Metal epoxy may be used to hold the screws in place and also to seal any mistakes.

The exhaust plate is cut and drilled in the same manner as the intake plate, except the center hole is small enough to allow only the exhaust pipe to fit through, but not the exhaust pipe *and* the four screws drilled into it. This means it is slightly larger than 1-3/8" in diameter.

Putting Things Together:

After the 3 of 5 total major components (intake port, combustion chamber, exhaust pipe, ignition system, and fuel system), are completed, they are ready to be assembled. Nothing is yet made too permanent, as it is possibly necessary to take everything apart several times to get it right. Four 12" lengths of 10x24 threaded rod (From the hardware store. They are 1/8" in diameter for those like me, who had no idea that a 10x24 threaded rod has approximately that diameter), are put through the four corner holes of the first two plates, with the gasket sandwiched in between The intake plate and valve plates are pressed together using the 4 threaded rods and a total of 8 nuts on the front of the intake plate and back of the valve plates. (The intake plate, valve plate, and gasket are now in intimate contact, and can be considered one piece, so the nuts are screwed on tightly.).

Four more nuts are threaded loosely over the rear end of the threaded rod, and moved several inches forward of the rear (narrow) end of the combustion chamber. Next, the combustion chamber's front (wide) end is centered over the valves and the exhaust pipe's flat end (with the screws) is slipped over the lip on the narrow end of the combustion chamber. The tail pipe should fit perfectly over the narrow mouth of the combustion chamber. The exhaust plate is then slipped over the exhaust pipe, and brought firmly against the four 90^{o} screws in the exhaust pipe.

The final four nuts are screwed on the 10x24 rod to tightly press from the rear the exhaust plate against the 4 screws in the exhaust pipe. The exhaust plate presses on the exhaust pipe screws, which press on the exhaust pipe, which in turn presses against the narrow end of the combustion chamber, whose wide end is in turn pressed against the valve plate. The valve plate is held to the intake plate not only by the pressure from the combustion chamber, but also by the 8 screws in the front of the assembly. Adjust the rear 4 nuts such that the exhaust pipe is in direct line with the combustion chamber. If the front end of the exhaust pipe was cut perfectly perpendicular (or was left as-is uncut from the factory), this adjustment should not be too difficult. The forward 4 nuts of the rearmost 8 (the ones that were brought forward several inches) are now brought rearward to be in contact with the exhaust plate. These are tightened as necessary. Make sure all nuts are on tightly and nothing slides around. The engine is now 75% assembled. The remaining parts are not necessarily attached to this combustion chamber/tail pipe assembly.

Place your mouth to the front of the engine and blow hard. There should be little or no restriction to the flow of air from the front to the back. Next, blow *backwards*, in from the exhaust pipe, There should be *no* airflow whatsoever. If

there is (and there probably will be) make whatever adjustments are necessary to the valves in front. This concludes the main engine section.

IGNITION SYSTEM:

The ignition system is used only initially to start the engine. After the engine has started, the ignition system is no longer needed, as further ignition is carried out by hot gases reentering through the exhaust pipe.

Anyone reading this packet should be familiar with a generic automotive ignition system. The original ignition system is based on a RS540 radio controlled model car motor to make and break contact so that the spark plug can work properly. The alternate ignition system described here is based not on a mechanical timing device (motor and battery) and a large contact switch (copper wires connected to the motor, and contacts on the base of the unit), like the original plans, but is based on a solid state timing device (555 timer) and a small contact switch (relay).

555 Timer ignition system:

PARTS -- all available from Radio Shack (catalog numbers are given for some parts) or other electronic parts store:

1 555 timer chip -- Cat. no. 276-178.
1 8 or 16 pin socket to fit 555 chip. Cut 16 pin socket in half to 8 pins.
1 47,000 (47k) ohm resistor -- less than $1 for a pack of 10.
1 100 ohm resistor
10 microfarad capacitor. -- bought in bulk (10 or more usually)
1 12 volt or greater, 3A relay. Cat. no. 275-248
1 12 volt hobby battery. (not available at Radio Shack)
1 1N4001 or equivalent diode. Cat. no. 275-1653
1 SPST switch, large enough to look hefty and strong and rated at least at a few amps.
Pre drilled electronics breadboard (with the copper around the holes) about 2" x 2". Make sure to get the kind for ICs. Cat. no. 276-150
Solder and iron
Some thin plastic insulated wire. 22-24 gauge.
Spark plug and appropriate plug cable
Automotive ignition coil (small as possible to keep budget low)

Theory:

The 555 chip is very general purpose timer/oscillator. With a few external parts (2 resistors and a capacitor) it can be configured to provide a repeating on-off signal ranging in time intervals (periods) from microseconds or less to several minutes or hours, depending on what the values of these parts are. The chip's pins are numbered starting with number 1 at the upper left corner, and continuing down and around to pin 8. "Upper" means the end with the notch or semicircular shape at that end, as is shown in the second diagram. The pins are NOT configured the way it is shown in the second diagram below. It is shown this way only for the purpose of clarifying the rest of the circuit. See the first diagram for what the pins really look like on the chip. The output of the chip on pin 3 goes from 12 volts to 0 volts repeatedly. The time per cycle which the output stays at 12 volts is determined by the formula

$$T_H = 0.693 \times (RA+RB) \times C,$$

where RA and RB are the 100 and 47,000 ohm resistors, respectively, and C is the value of the capacitor in farads, 0.000010F, or 10×10^{-6}F. The value for time T_H in this case works out to 0.326 seconds.

The time per cycle the signal spends at 0 volts is determined by the formula

$$T_L = 0.693 \times RB \times C.$$

The value T_L in this case also works out to 0.326 seconds. These values are rounded off to 3 decimal places. Therefore, the period, or total time it takes for the chip to complete one cycle, or one length of time at 12 volts and one length of time at 0 volts is the sum of the high and low states,

$$T_L + T_H$$

For this case the total period T works out to 0.652 seconds and the frequency or cycles per second is $1/T=1.533\ s^{-1}$. It is necessary to have several cycles (sparks on the spark plug) per second in order to assure a near continuous heat source for when the engine is started. If it is necessary to change the timing of the spark plug, make the necessary adjustments to either resistor(s) or capacitor, or all three.

If you are not familiar with simple electronic schematics, following is the basic concept. A schematic is a *symbolic* representation of certain parts of a circuit. The actual working circuit with real parts is not supposed to look at all like the

schematic. Each line from one part to another represents a wire which has been soldered between those two parts at their respective locations. A dot where two or more lines meet indicates that these wires all carry the same connection. As you can see almost all wires in the schematic eventually lead to what is known as "ground." It really has nothing to do with the earth, but that is the convention given. The symbol for ground is several horizontal lines decreasing in size. Practically, ground is used because every current eventually needs to find its way back to the negative end of the battery. That is why the negative end of the battery also carries the ground symbol. Even though the battery ground and the ground near the chip and ignition coil are connected to separate ground symbols in the schematic, it is the same connection in the real-world circuit. In real life ground is basically a bunch of melted solder on the bottom of the circuit board which connects all the appropriate points to the negative end of the battery.

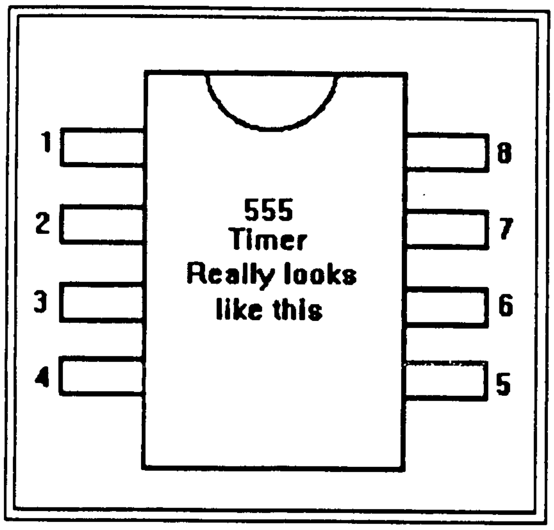
The schematic for the relay is shown on the back of the packaging in which the relay came. It should be similar to the one shown below. Basically the circuit works like this: The 555 timer sends a stream of on-off pulses to the relay. The relay is a switch, like what controls the lights in your house, except instead of a hand or a set of fingers flipping the switch, it's done by a small electromagnet inside the relay. The signal from the 555 is what turns the magnet on and off.. The schematic on the back of the packaging makes it rather clear what's going on inside the relay.

Once the switch inside the relay is closed (so that current flows from the battery to the positive side of the ignition coil, and out the negative end to the ground, which is the negative end of the battery), a magnetic field is set up inside the ignition coil. It is not necessary to understand what this is. Just know that when this current to the ignition coil is shut off, the magnetic field inside the coil collapses and sets up a very high voltage, on the order of thousands of volts, across the spark plug. This high voltage is enough to rip apart (ionize) the air molecules, and cause a nice hot spark. The diode allows current to flow only away from the 555, so that it doesn't get damaged from high voltage from the spark plug trying to get back into the chip. Adjust the gap in the plug for the biggest, as well as most consistent spark. Be sure that the side of the spark plug is connected cleanly somehow to ground, and that you are not touching the plug when the thing is turned on, because you may get a harmless but sobering jolt.

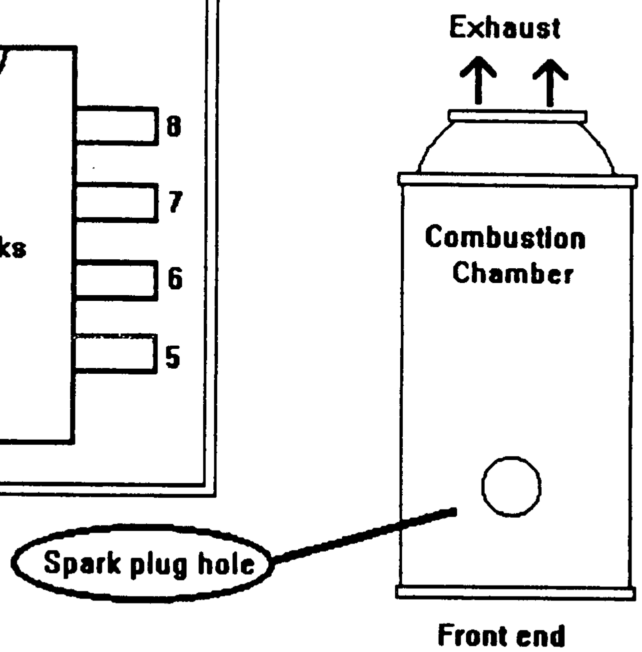
Basic soldering techniques:

First determine the basic layout of the parts on the circuit board. The components themselves should be on the non-copper side, with all the connections, made with solder, on the copper side of the board. The layout is not crucial, but just be sure you don't get confused about what goes where when you flip the board over to work on the other side.

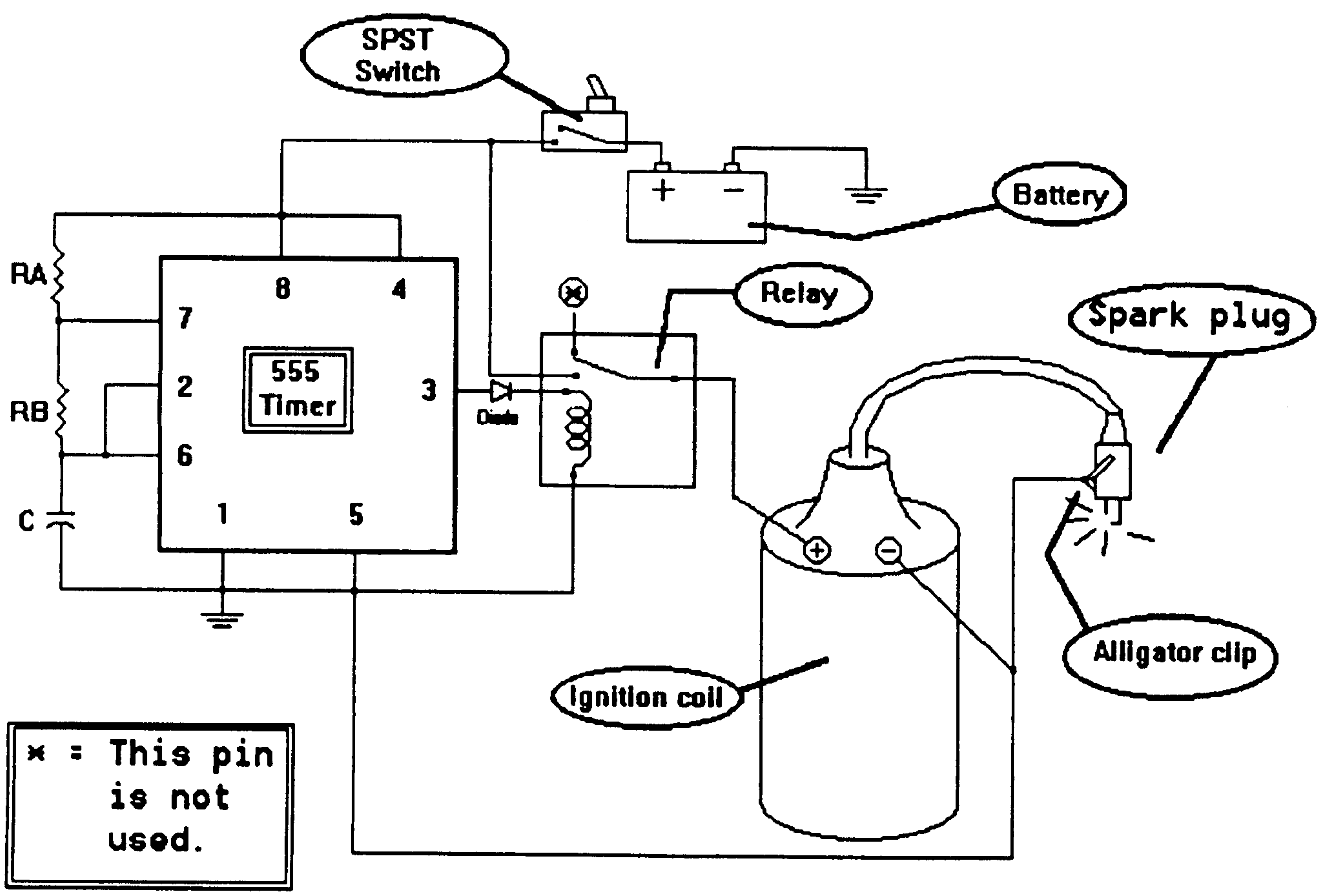
Your soldering iron should be rated at no more than 30 or so watts for electronic board soldering. Plug it in and let it heat up all the way. First determine which parts are to be heated. Then apply as much of the tip as you can to both parts to be heated. Do not add solder yet. After several second of heating the parts, touch the end of the solder to the hot *part,* not the soldering iron itself. The solder should flow smoothly. Remove the iron from the connection and let the solder cool. A good connection is solid, smooth, and shiny, not loose, dull, and gray. Use only enough to hold the wire in place firmly. Do not glob it all on. You will know when you make a bad connection -- it just doesn't look right. Soldering is not very difficult after a bit of practice



1
2
3
4
555
Timer
Really looks
like this
8
7
6
5



Exhaust
Combustion
Chamber
Spark plug hole
Front end



SPST
Switch
Battery
RA
RB
C
8
4
7
2
6
1
5
3
555
Timer
Relay
Spark plug
Alligator clip
Ignition coil
× = This pin
is not
used.

Board assembly:

Start first by soldering the resistors, as these are the components that protrude the least from the top of the board, and are easier to solder by laying the board upside down on the table, and letting the resistors come into complete contact with the board. The ground along the back side of the board, as mentioned earlier, is a stream of solder that runs the length of the board end to end. This is so that as many connections as possible can be made to it.

Next solder in place the 555 socket. If you choose not to use the socket and to connect the 555 directly to the board, do the soldering quickly, so that there is no heat damage to the chip. After this add the capacitor making sure to observe its polarity if you are using an electrolytic capacitor. You may need to add some wire here and there to make everything connect properly. And finally, add the relay..

After all the parts are in place, add the proper ground wire to the ground mess on the back side of the board, and the wires from the positive end of the battery to the board and proper place on the relay. Make CERTAIN that the wire to the relay from the battery is attached to the pin that is normally OPEN. This is seen clearly in the schematic below as well as in the schematic on the relay packaging. If this is done improperly, the battery will be drained whether or not the switch is open. Don't forget to insert the 555 into the socket after the socket is in place. Also, don't forget to include the switch between the + end of the battery and everything else, like in the schematic. Use an alligator clip or something on the spark plug which is connected to ground (-) somehow, so that the plug is easily detached from the ignition electronics. Also, make sure the diode has the proper polarity. The white line or mark on the end of the diode corresponds to the "forward" tip of the symbol in the schematic, the end pointing at the relay.

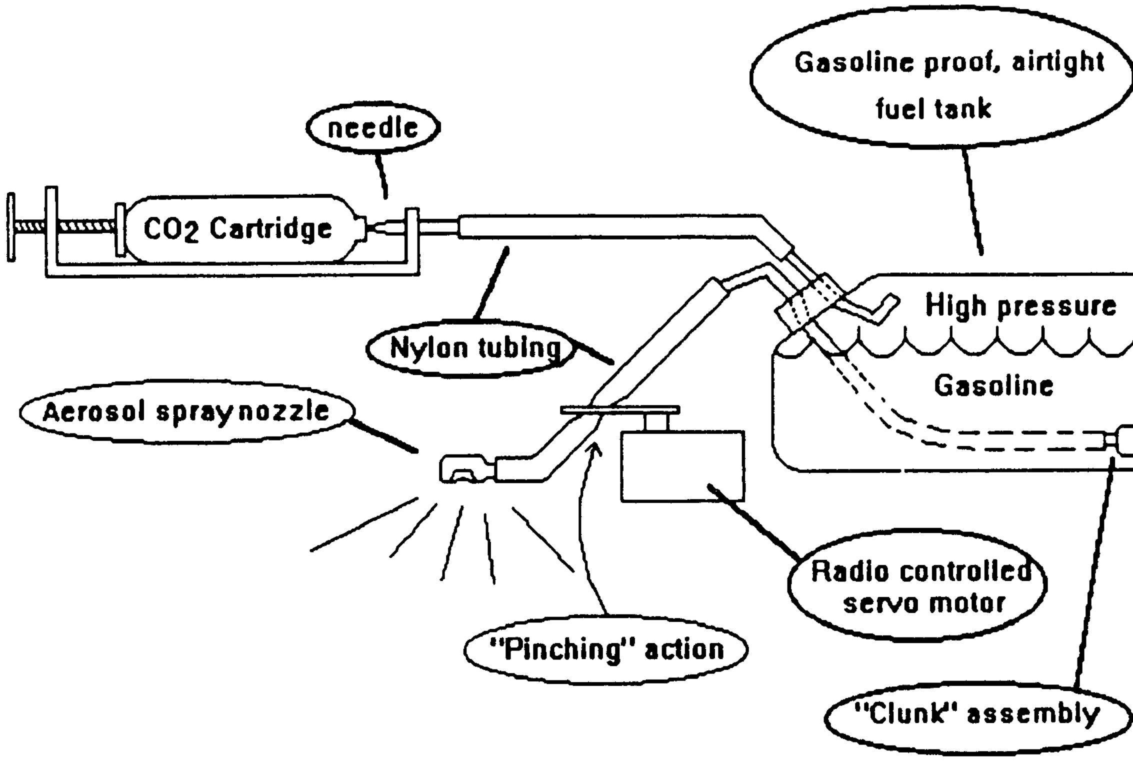
Turn the thing on, and there should be a clicking sound coming from the relay, and the spark plug should be sparking. Assuming you've read these instructions before actually building anything, you'll know to test the spark plug and ignition system before assembling everything! If the chip should somehow blow up, or start smoking, or fizzing, turn off the circuit, remove the now bad chip, and check everything to make sure there are no solder connections where there shouldn't be, and everything is as neat as possible. If you still can't figure what's wrong, see if there's a person where you got all your supplies, and maybe he or she can help you.

FUEL SYSTEM:

The original fuel delivery system in the plans from *Jet Engine Technologies* is based on a 12 volt on board battery and a small fuel pump. This is not only heavy, but basically limits the engine to either on or off. If the builder were to install a variable flow gasoline system using this battery/pump method, he would need to use a variable voltage source to the fuel pump, such as an R/C car speed controller. Suddenly, the model plane is not so light, and simple.

The author's idea is based upon using a conventional model airplane fuel tank, *made specifically to hold gasoline.* (Gasoline will dissolve fuel tanks made for normal nitromethane based model airplane fuel.) This airtight tank is pressurized by a small, disposable CO_2 cartridge normally used for paint guns or BB guns. Set up the fuel tank according to the instructions in a "clunk" configuration. This assures that the tank can be used in any physical orientation, and fuel will always find a way to the engine, no matter in what position the airplane is.

The needle/screw assembly normally found in these paint or BB guns is used, and mounted in the airplane, or wherever the engine is being tested. Nylon or vinyl tubing is used to connect the CO_2 cartridge to the fuel filler tube on the fuel tank. The output tube is connected to an aerosol can spray nozzle. The spray nozzle must put out a very fine mist of fuel, and must spray directly into the center of the intake manifold. Practice with water first. A radio controlled servo motor and pieces of wood or plastic are fashioned in such a way that the servo can pinch the tube either going into the tank to cut off the pressure to the tank, or pinch off the fuel itself coming from the tank. This method will allow a direct control of fuel, with no dangerous (sparks) battery or expensive fuel pump. See diagram. The spray nozzle is epoxied to piece of scrap aluminum which is itself epoxied to the outer intake cone of the engine.



STARTING:

To start the engine first make sure the fuel pump sprays properly and the ignition system is running properly. Somehow get a stream of air going into the front of the engine. The original plans say that a blow dryer blows enough air to get the engine going. It is difficult to imagine running a blow dryer at the flying field. The following is a suggestion as to how to get a proper initial airflow into the pulsejet engine.

Since the modeller probably already has built several flying airplanes by the time he is reading this information packet, he probably has a good idea of how to mount a normal single piston model airplane engine into some sort of a wood mount. A wood or glass filled normal-engine motor mount screwed to an aluminum house-gutter-aluminum funnel shaped tube would allow a normal engine fitted with a small propeller to blow directed air back into the pulsejet engine. The narrow end of this sheet aluminum funnel would need to be larger than the opening of the outer intake cone, and the large end of the funnel would need to be larger than the diameter of the propeller mounted on the normal piston engine. This is a round-about way of delivering air to the pulsejet engine, but it may work.

After the air is flowing properly, start the ignition system. Once the spark plug is running properly, *slowly* turn on the fuel spray. Make sure it sprays directly into the center of the intake manifold. Eventually the fuel should ignite with a loud bang. Keep trying until it catches by itself. When the engine is running on its own, it will pulse around a hundred or so times per second. Keep all body parts away from the engine until you are familiar with how it runs.

Perhaps some additions can be made to the front of the intake manifold so that the amount of air can be adjusted via a choke or vanes controlled by the same R/C signal which controls the rate of gasoline to the intake. This, however, would have to be calibrated very precisely, and would be rather difficult to implement into this simple engine.

HAPPY MODELING!

Please send any comments, complaints, or suggestions for improvements, to:

MASTECH
830 Child's Way Suite No. 164
Los Angeles, CA 90089

Original plans may be purchased from *Jet Engine Technologies* for $12.95 for the large size (7lb thrust) model, and $9.95 for the smaller (3lb thrust) model. Both can be purchased for $20.00. Prices are good as of February, 1993. Check with them to be sure.

The address:

Jet Engine Technologies
10241 Ridgewood
El Paso, TX 79925